POWER

Volume 58 New York, August 21, 1923 *Number* 8

A Purpose for the Power Show

THE Power Show held at New York last December, following the meeting of the American Society of Mechanical Engineers, was a decided success, and that being planned for next December promises to excel the first in the number and interest of the exhibits.

With the dignity consonant with a learned professional body, the American Society of Mechanical Engineers has carefully guarded its meetings and proceedings from any taint of commercialism. He who installs an exhibit even in the official hotel is quietly told that it is distasteful. A hustling promoter who got into the hall before a meeting and placed a sample of his literature in each of the seats was followed by an attendant who gathered it up and returned it to him after the session.

But while properly jealous of any perversion of its own prestige and facilities to other than unbiased professional purposes, the Society is not only tolerant of, but actively interested in, the exhibition of mechanical and power-plant apparatus that offers to its members and the attendants at its meetings an ocular demonstration of the latest developments in the field and a chance to compare the appliances and materials available for power-plant purposes.

And the thought occurs that the Power Show might perhaps, without offense to the dignity of the Society, be linked more closely with its meetings by furnishing an exemplification of some of the things, capable of physical demonstration, treated of in the papers.

When Mr. Arkimof, for example, read his paper descriptive of his apparatus for balancing rotating parts, how interesting and clarifying it would have been, and how conducive to a broader understanding of the principles and possibilities of the system, if one could have, before or after the presentation of the paper, seen the operation of balancing a big turbine rotor or crankshaft.

How much light would be thrown upon and interest kindled in a paper upon nozzles or orifices by an exhibit showing their effect as did the Bailey exhibit of last year.

It would seem that there might be many demonstrations, impracticable or undesirable upon the lecture platform, which would make added attractions for the Power Show and help in making the papers which they illustrate more widely discussed and intelligently understood.

F. R. Low

Some Instrument Practices of the United Electric Light & Power Company

By W. E. CALDWELL
Assistant to the General Superintendent of Power Plants

THIS article explains how the United Electric Light & Power Co., which operates the huge Sherman Creek and Hell Gate Stations in New York City, safeguards the accuracy of its fundamental test data. Complete instructions are given for the construction of a precision mercury vacuum gage, and for testing steam gages in position. The comments on thermometer selection are also of practical value.

ONE of the most important records in connection with either condenser or turbine tests is that of vacuum or absolute exhaust pressure. The usual mercury-column vacuum gage, while more or less satisfactory for routine operating records, is woefully inadequate for conducting tests requiring a high degree of accuracy. The chief structural sources of error with the conventional mercury column are due to the small-bore tubing usually employed, coarse graduations, indefinite zero adjustment, and most important of all is the absence of any positive means of calibration. The ordinary mercury absolute-pressure gages are open to the same objections, and owing to the principle of their operation the absence of positive means of calibration is even more serious than in the former type. Aside from the structural deficiencies of these instruments the difficulty of accurate reading also exists.

Some years ago the writer, after experiencing difficulty in securing consistency in turbine-test results, devised a mercury column that seems to overcome the deficiencies of the conventional type and, incidentally, at a lesser cost. The neat appearance and precision of the Tycos mercurial barometer indicated possibilities that led to the adaption of a barometer case for the purpose of measuring vacuum.

A barometer, without the mercury tube, but otherwise complete with mercury reservoir, zero pointer, vernier, etc., was obtained, and with but few modifications was easily adapted for measuring vacuum. The internal stuffing box for the top support of the glass tube was removed, together with the cap containing the supporting hook. The supporting hook was removed from the cap and the latter was drilled to receive a piece of ¼-in. brass pipe. A ¼-in. brass tee plugged at one end was provided with ¼-in. brass nipples about an inch long screwed into the side and bottom of the tee with the bottom nipple extending through the cap for connecting to the glass mercury tube. A small collar on the lower nipple fits against the cap for supporting the case, and a small hook attached to the plug in the top of the tee completes the supporting arrangement.

The mercury tube is made of lead-free glass about 33 in. in length, open at both ends, having a bore of ⅛ in. from the bottom to about the 26-in. mark and a bore of ¼ in. for the remainder of its length. It is connected to the brass nipple below the cap by means of a short piece of rubber tubing which insures an airtight flexible joint. The lower end of the tube extends below the surface of the mercury in the reservoir. In order to increase the range of zero adjustment of the mercury reservoir, the leather packing in the bottom recess within the reservoir was removed, permitting it to be raised higher before coming in contact with the bottom of the tube.

The transformation of the barometer into a mercury column having been completed, a simple and accurate means of calibration was necessary. The problem was simply to measure, accurately to a thousandth of an inch, the distance between a horizontal plane touching the bottom of the zero pointer and one passing through the 30-in. graduation on the scale. For this purpose a ¼-in. gage rod of steel ground to within a thousandth inch of 30 in. at 62 deg. was made to replace the glass tube during calibration. An adjustable parallel-face gage of steel with polished upper surface was fashioned so as to fit snugly into the chamber ordinarily occupied by the mercury reservoir. The upper surface of this gage may be regarded as an artificial mercury level. The procedure for calibrating the column is as follows: The glass tube and reservoir are removed from the case, the gage rod is substituted for the tube, while the gage with parallel faces replaces the reservoir. The artificial mercury level is adjusted to make just perceptible contact with the ivory pointer while the lower end of the gage rod rests upon this same artificial level. The vernier at the top of the column is set exactly even with the top of the gage rod and then locked. If the 30-in. mark on the scale does not coincide with the upper surface of the gage rod, the scale is then adjusted accordingly.

In order to obviate the necessity of any correction for difference in expansion between the steel gage rod and brass case, the calibration is carried out at a temperature as near 62 deg. F. as possible. Since the scale of the ordinary barometer, to which it is necessary to correct the indications of the mercury column, is calibrated for the same temperature, this places both instruments on a common basis and simplifies the calculations required in referring the indications to the standard temperature. When the calibration is effected at a temperature other than 62 deg., the necessary slight correction is calculated and applied by shifting the scale on the column accordingly.

At *F* in Fig. 1 may be seen the cylindrical case of an instrument with mercury tube and reservoir removed preparatory to calibration. The mercury tube system and mercury reservoir are shown at *D* and *E* respectively. In order to render the photograph more distinct, the mercury tube was filled with colored liquid which imparted conspicuousness to the bore of the tube, especially at the enlarged section. At *B* and *C* are shown the 30-in. gage rod and the zero gage or artificial

mercury level used in connection with the rod in calibrating the instruments. The assembled column appears at *A*.

The calibration of the column may be effected with great precision since the vernier in connection with the graduated scale permits of accurate measurements to the thousandth part of an inch. In actual practice the indications of vacuum are recorded only to the nearest hundredth of an inch, but it is, of course, desirable to secure the highest reasonable instrument accuracy.

After calibration the reservoir is filled with mercury and the instrument re-assembled, the entire procedure of dismantling, calibration and assembling requiring about ten minutes. Pure mercury is always employed. Owing to its peculiar properties the determination of density is an unnecessary refinement, even assuming that the problem were as simple as it might appear to one unfamiliar with the real difficulties involved. Fortunately, mercury contaminated with sufficient lead, tin or other element to produce an error of one-thousandth of an inch of vacuum, due to change in density, is so foul that it is rendered unfit for use in a mercury column. When mercury containing the most minute traces of impurities is employed, a dark coating is quickly deposited within the tube which soon entirely obscures the meniscus. Such a coating furnishes ocular evidence of contamination of the mercury, the continued use of which would result in errors of observation far greater than any that could possibly be produced by the accompanying reduction in density. In passing it may be stated that whenever a test is reported in which corrections have been made for density of mercury, other than that due to temperature, it would appear to imply a degree of accuracy which is simply impossible of attainment, to say the least.

Three columns such as described check so closely when connected to a common line communicating with a surface condenser as to leave little doubt as to their relative accuracy. When connected in this manner, their indications all agree within one-hundredth of an inch

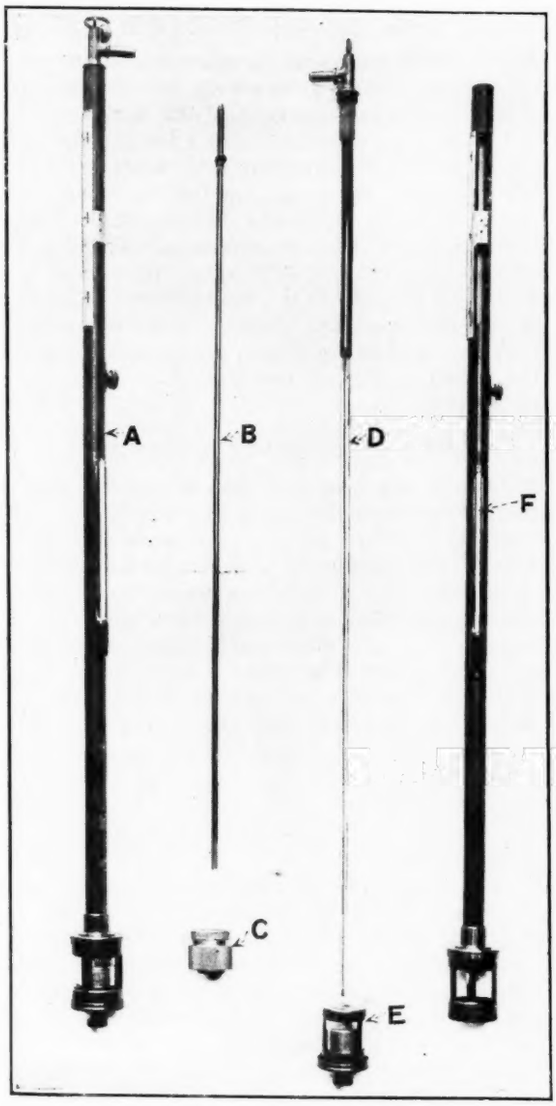


FIG. 1—SHOWING A VACUUM GAGE MADE FROM A BAROMETER

A—Finished vacuum gage. *B* and *C*—Special gages used for setting scale. *D* and *E*—Mercury tube system and mercury reservoir. *F*—Showing the barometer partly dismantled to make gage.

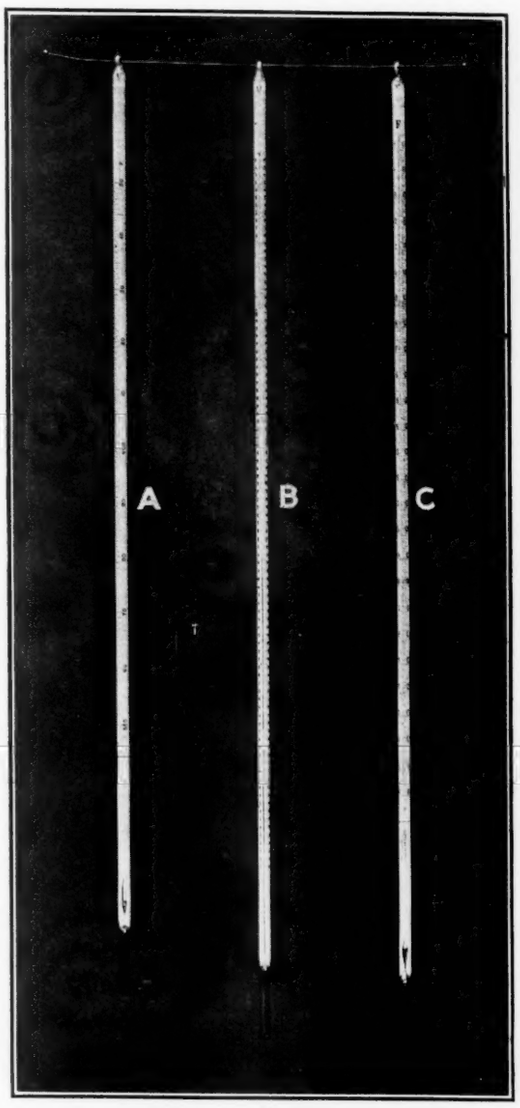


FIG. 2—CLOSE-READING THERMOMETERS FOR POWER-PLANT WORK

A—Superheated steam, 550 to 650 deg., 20-in. scale, calibrated 5-in. immersion. *B*—Condenser tests, 30 to 120 deg., total immersion. *C*—Saturated and superheated steam, 400 to 650 deg., calibrated 4-in. immersion.

if corrected to a common temperature. Taken alone, this test indicates nothing that would not be expected, but to one familiar with the discrepancies frequently observed between the indications of instruments of the conventional type under similar conditions, the close agreement shown by this simple test has its significance.

In addition to the calibration of these columns prior to conducting condenser or turbine tests, it is of course necessary to apply the usual corrections to the observed reading for referring the results to standard conditions. The first correction applied is for temperature, in order to reduce the observed indications from the existing temperature to the standard of 58.4 deg. corresponding with the mean barometer of 30 in. at sea level. Each

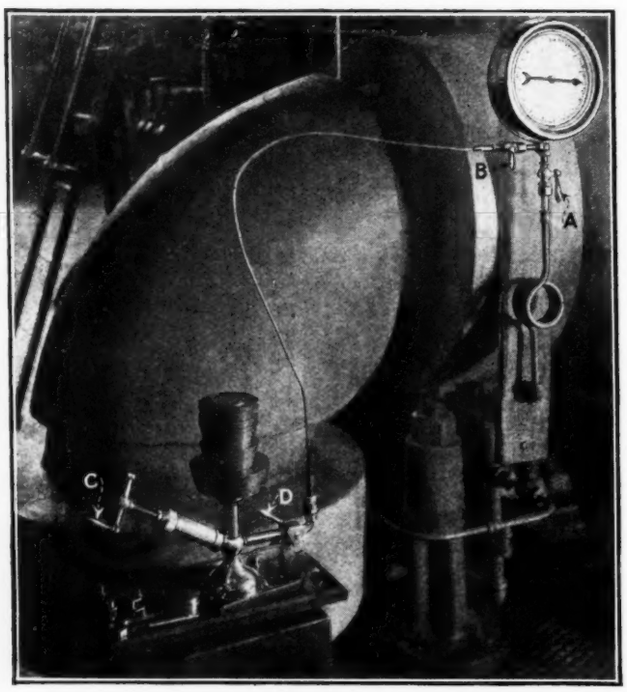


FIG. 3—ACCURATE AND CONVENIENT GAGE-TESTING SET-UP

instrument is provided with an integral thermometer, and by its indications the correction is obtained from physical tables. The correction for capillarity is next applied by reference to the physical tables. This correction depends solely on the bore of the tube, which is known, and the height of the meniscus, which may be accurately determined by means of the vernier. The height of the meniscus is usually about three-hundredths of an inch, hence the correction to be added, with the quarter-inch tube, is approximately two-hundredths of an inch.

The barometer is likewise corrected to standard conditions and, in order to simplify matters during a test, it is hung adjacent to the column measuring vacuum at the turbine exhaust, resulting in a common temperature and elevation for both instruments. For columns located in the basement to measure vacuum at the hotwell and air offtake, it is necessary to refer the corrected barometer reading to this elevation in order that all readings may be comparable. This correction for elevation might seem superficial, but actually is too important to neglect. For instance, in Hell Gate Station, when conducting tests, the column measuring vacuum at the turbine exhaust flange is at an elevation of 32 ft. above those connected to the hotwell and air offtake, hence at the temperature usually prevailing there is a difference of atmospheric pressure of about three-hundredths of an inch between these two elevations.

The corrections to mercury columns and barometer having been effected, the final step of converting the results into terms of absolute pressure is completed by deducting the corrected vacuum from the corresponding corrected barometer reading.

It may be observed from the foregoing that the accuracy of the measured vacuum or exhaust pressure, after applying all the necessary corrections, is still contingent upon the accuracy of the barometer which, unfortunately, cannot be independently standardized. However, it may be easily compared with the local U. S. Weather Bureau barometer, and if the comparison is favorable all doubt is removed. For this test it is necessary to correct the barometer reading to 32 deg. F. and mean sea level, since the Weather Bureau refers the observed reading to this standard before reporting. With stable atmospheric conditions the writer has had no difficulty in checking within one-hundredth of an inch of the Weather Bureau record, a barometer some ten miles removed.

Special Thermometers Employed

For measuring temperatures during tests as well as in regular operation, it has long been the practice of this company to employ thermometers especially adapted to the purpose in order to secure the highest possible accuracy and sensitiveness. The temperature of steam to a turbine as affecting its economy is secondary only to that of vacuum, and the accurate determination of the steam temperature is therefore of great importance. For this purpose thermometers are employed having a total scale range of but 100 deg. graduated from 550 to 650 deg. F., in a total length of 20 in. and calibrated for 5 in. immersion. These instruments may be easily read to a quarter of a degree, and the correction for stem exposure is practically negligible.

The magnitude of the stem correction encountered when using the conventional type of thermometer graduated from 0 to 650 deg. and scaled for full immersion, may be appreciated upon referring to an article on this subject which appeared in the Feb. 27 issue. Under ordinary conditions, when measuring the temperature of steam at 600 deg. or above the stem correction amounts to around 20 deg. to be added. With the type of thermometer employed by this company there is but 50 deg. of emergent stem when measuring temperatures of 600 deg., and the temperature of the exposed portion is usually within 25 deg. of that for which it was calibrated, so the correction for stem exposure seldom exceeds one-tenth degree and may consequently be omitted.

The steel wells used with these thermometers have sufficient depth so that the immersion mark is always covered with mercury. For the slight evaporation that takes place, more mercury is added as required. Conjecturally, a steel well screwed into a pipe carrying superheated steam would be subjected to a temperature somewhere between that of the steam and that of the pipe, but even this assumption is not as serious as it might appear. With the usual steam velocities of around 10,000 ft. per min. and the resulting turbulence, together with fairly efficient pipe covering, the tem-

perature of the pipe is in all probability exceedingly close to that of the steam. The writer is convinced that where reasonable care is exercised, the temperature of high-pressure superheated steam may be determined with a high degree of accuracy, popular conception to the contrary notwithstanding.

Other thermometers with suppressed scales similar to the foregoing are employed for various other temperature measurements about the plants, and graduated generally for the specific application. With about four scale ranges to choose from, it is possible to select instruments best adapted for the various measurements incidental to any test. Quite aside from the accuracy and sensitiveness of these instruments, the ease of observation is greatly increased. For condenser testing, where ordinary temperatures prevail, no scale suppression is required. For this purpose full-immersion precision-type thermometers 21 in. in length, graduated in tenths from 30 deg. to 120 deg., are employed. This range fully covers such requirements as turbine exhaust-steam temperatures, hotwell temperatures, circulating-water inlet, outlet temperatures, etc., for both summer and winter conditions.

Fig. 2 shows three types of thermometers employed for the more important temperature measurements incidental to the usual tests. Thermometer *A* is graduated from 550 deg. to 650 deg. for indicating the temperature of superheated steam. Thermometer *C* is also for superheated steam, but is graduated from 400 deg. to 650 deg. in half degrees, covering the entire range from saturation to the maximum steam temperature encountered at Hell Gate Station. Thermometer *B* is of the full-immersion type and graduated from 30 deg. F. to 120 deg. F. in tenths. This is employed for temperature measurements in connection with condenser or other tests for which the range is best adapted. The instruments illustrated, owing to their length and delicacy, require careful handling and are intrusted only to skilled observers.

Measuring Steam Pressures During Tests

Fig. 3 illustrates the practice followed in measuring steam pressures during turbine tests. As may be observed from the illustration, the dead-weight gage tester is connected in such manner that the gage, measuring pressure adjacent to the throttle, may be frequently tested without the necessity of removing the gage or releasing the pressure. The connection is simplified and rendered flexible by the use of small-bore copper tubing. The usual method of gage testing is laborious, and the results obtained are in many instances of doubtful accuracy. By the common method of standardizing it is necessary to remove the gage from the point of application in order to connect with the dead-weight tester. The necessary handling of the gage is not conducive to maintaining accuracy, especially with a class of instruments sufficiently sensitive for this work. The manner of releasing or restoring the pressure to the gage mechanism often results in errors that cannot be detected without removing the gage and rechecking. Failure of subsequent tests of the gage to check with the initial calibration invites speculation as to which set of corrections should be applied. These difficulties are overcome by means of the connection shown in the photograph.

In testing the gage, cock *A* is closed, after which cock *B*, communicating with the tester, is opened, the proper number of weights having previously been added to the tray and the handwheel *C* having been screwed down to insure full pressure in the connecting pipe. By this means there is not sufficient change of pressure within the gage mechanism to produce shock, and the gage may in fact be regarded as an intermediary through which the steam pressure is balanced against the dead-weight tester. Water head in the connecting piping is compensated for by a fixed correction depending on the piping arrangement. Although the desire for accuracy was the consideration that prompted this simple expedient, greater convenience was also secured, since the observer is able to check the gage between tests or even between observations when indications suggest the necessity.

Fusing Potential Transformers

By C. M. King

Potential transformers are employed to insulate electrical instruments and relays from high-voltage circuits and to avoid the use of resistances of undesirably high values, the secondary voltage of these transformers being approximately 100 volts. Since potential transformers are commonly used to energize important devices, such as the alternating-current control magnets of generator-voltage regulators and voltage relays, the incorrect functioning of which would probably cause serious operating difficulties, in some stations protective fuses are omitted, the transformers being connected directly to the busbars. Fuses are omitted because it has been found that, if of the proper current-carrying capacity to really afford protection, they sometimes gradually deteriorate in service and finally rupture the circuit without apparent cause. It is probable that this deterioration is the result of a static discharge gradually corroding the small conductor required and the constant vibration to which such fuses are frequently subjected, tending to bring about mechanical failure.

Potential transformers having a rating greater than 200 volt-amperes are seldom used, and the full-load primary current of a 11,000-volt transformer rated at 200 volt-amperes would be approximately 0.02 ampere. The short-circuit impedance of a potential transformer is low, and if only 2 per cent, with the secondary terminals short-circuited, the primary current would be but 1 ampere. Therefore, if a fuse having a rating in excess of 1 ampere was connected in series with such a 200-volt-ampere transformer, absolutely no protection would be afforded against trouble developing beyond the secondary terminals. Since high-voltage fuses rated at 1 ampere have been known to fail without apparent cause, under ordinary conditions of operation, it is necessary to use a fuse rated too high to provide adequate protection, to renew the fuses frequently, or to eliminate them entirely.

If the rating of the fuse is too great or if fuses are omitted, a short-circuit in the secondary wiring will damage the transformer winding because of excessive heating, and when failure of the insulation results, the short-circuit current may destroy the fuses if they are provided, or the primary leads may be destroyed. In either case the arcing, which is likely to develop between phases or between one phase and the ground, may cause a serious disturbance or a total interruption to service. Hence, if immunity from unnecessary failure of voltage is sought by using primary fuses of excessive rating or by their entire omission, not only

is the transformer without adequate protection, but serious trouble will probably develop if the transformer fails.

In some large plants resistance of low value is connected in series with each primary fuse, this serving to reduce the short-circuit current, under the worst conditions, to a value that the fuse can safely rupture. If a 10-ohm resistor is connected in series with each of the two fuses required to protect a single-phase 200-volt-ampere 11,000-volt transformer, the total resistance in series with the primary winding will be 20 ohms and the maximum short-circuit current would be about 550 amperes. Since at full load a resistance of 20 ohms occasions a loss of but 0.008 watt and a drop of but 0.4 volt, its effect on the secondary side is negligible.

Although fuses are on the market which will effectively isolate a 200-volt-ampere transformer if the secondary terminals are short-circuited, the scheme of placing fuses in the secondary wiring as near the transformer as possible offers certain advantages, and this method of protection is used in some stations. By selecting fuses on the basis of the actual secondary load, the transformer can be well protected against trouble occurring in the secondary wiring. The primary fuses, which should be connected in series with suitable resistors, would function only in event of trouble in the transformer and could be safely rated high enough to possess ample mechanical strength.

Drum Suspension for Boilers

BY J. B. CRANE

There are a great many ways of supporting drums of vertical, semi-vertical and cross-drum boilers. Some of the common methods are shown in the illustration.

In Fig. 1 the drum is shown extended and resting on the side beam connecting the columns. This is probably the oldest and simplest and until recent years was in greatest use. It has the following disadvantages: (a) Hard to level the drums if the two side beams are not level; (b) the joint between the drum and brickwork is difficult to keep tight, especially if the drum is near the stack uptake when the draft is high; (c) no means are provided for expansion and contraction, and in long drums they must slide over the side beams or make the side beams move in or out with consequent brickwork trouble, as their side beams usually act as buckstays for the brickwork; (d) leaves more drum surface to be covered with insulating material to prevent radiation; (e) in battery settings more space is required and a very thick party wall.

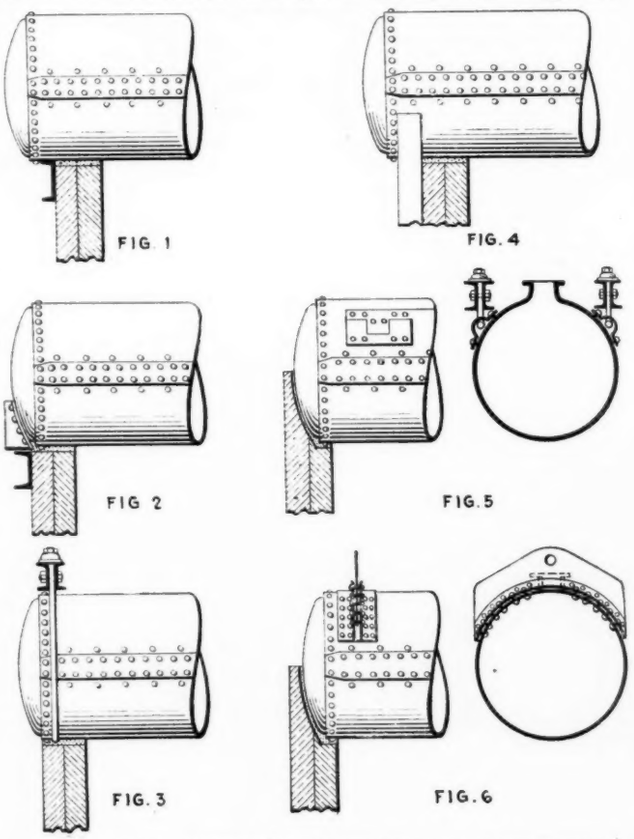
The drum in Fig. 2 illustrates a common method of support now used. At higher pressures and with thick plates in the drums it is cheaper than the method of support shown in Fig. 1. A lug is riveted on each head, and this lug in turn rests on the side beams. The stresses on the heads are not capable of exact computation, and it is customary to build them ½ in. thicker than would otherwise be necessary in order to take care of this unknown quantity.

It is superior to the first one in that it can be leveled easier by putting shims under the lugs; there is not quite so much drum surface exposed to the radiation, and it does not require so much space and such thick party walls for battery settings. There is still no provision for expansion and contraction, and the joint between drum and brickwork is difficult to keep tight.

In Fig. 3 is illustrated the familiar support with U-bolts. This provides for expansion and contraction and requires a drum longer than that shown in Fig. 2, but not as long as that shown in Fig. 1. Otherwise, it has the same objections as the other two and requires more height than the other two.

Fig. 4 shows the drum supported by a channel run from the floor. The channels at both sides of the boiler are also tied together by other channels in addition to the drum. This method of suspension has been used only in one installation so far as I know, and has nothing of advantage over the other methods shown.

There seems to be a growing tendency to support the drums as illustrated in Figs. 5 and 6, where lugs are



FIGS. 1 TO 6—DIFFERENT METHODS OF SUPPORTING BOILER DRUMS

riveted to the drums themselves and these lugs supported overhead. In the method shown in Fig. 5 the overhead support is no higher than the steam nozzle. This support is capable of exact computation. Most of the rivets are in shear and stressed only one to two thousand pounds per square inch. The drums can be leveled easily, and expansion and contraction are provided for. In battery settings the space is at a minimum and the party wall can be made as thin as desired.

The seal at the end of the drum between the drum and brickwork is much superior to the other methods of support, in that the drum is entirely inside of the brickwork. This method of support has been in use on a large number of boilers for several years, and not a single case of trouble has developed.

The method of support shown in Fig. 6 has been used where it is necessary to support heavier loads than that shown in Fig. 5.

Test of Diesel Fuel Consumption

By W. B. GREGORY*

A MUNICIPAL power plant was installed at Crowley, La., during the year 1922, to furnish electrical energy for street lighting as well as commercial lighting and power. The equipment consists of two Fulton four-stroke-cycle Diesel engines direct connected to General Electric three-phase 60-cycle 2,300-volt generators. The larger engine is a four-cylinder unit, rated at 380 b.hp. at 200 r.p.m., the generator being rated at 312 kva. The smaller is a three-cylinder unit developing 285 b.hp. at 200 r.p.m., while the generator is a 250-kva. machine. The plant is shown in Fig. 1.

The test was undertaken for the City of Crowley to determine whether the guarantees made by the Fulton Iron Works had been met.

Guarantee. The engine builder guaranteed the engines, when completely erected and operated, to have a consumption of fuel not to exceed the following:

100 per cent load.....	6.00 gal. per 100 b.hp.-hr.,	or .44	lb. per b.hp.-hr.
75 per cent load.....	6.50 gal. per 100 b.hp.-hr.,	or .478	lb. per b.hp.-hr.
50 per cent load.....	7.25 gal. per 100 b.hp.-hr.,	or .530	lb. per b.hp.-hr.
25 per cent load.....	10.00 gal. per 100 b.hp.-hr.,	or .733	lb. per b.hp.-hr.

These guarantees were based on fuel having a heat value of 18,500 B.t.u. and a gravity of 18 to 40 deg. Baumé at 60 deg. F., fuel oil to be reasonably free from foreign matter and to contain not to exceed 2 per cent sulphur, ½ per cent water and not over a trace of acid. The vender further guaranteed that the engines would not vary in speed exceeding 3 per cent from quarter to full load either way from mean speed, with gradual change of load, and that the engines would operate in parallel when generators were provided with suitable damping devices.

The tests were started Sept. 6, 1922, but were discontinued because it was found desirable to make certain changes in the fuel nozzles of the four-cylinder engine. The full load on the 285-hp. unit was run Sept. 6, 1922, and the other runs were made on Oct. 7 and 8.

The fuel oil used during the Sept. 6 test was Elm Grove crude having a specific gravity of 0.876 at 60 deg. F., equivalent to 30.05 deg. Baumé. For the later tests the fuel was Jennings crude with a specific gravity of 0.925 at 60 deg. F., equivalent to 21.45 deg. Baumé.

During the test fuel oil was carefully weighed in 15-lb. quantities and the time required to consume each amount was noted. A continuous speed counter was

*Prof. Mech. Eng., the Tulane University of Louisiana.

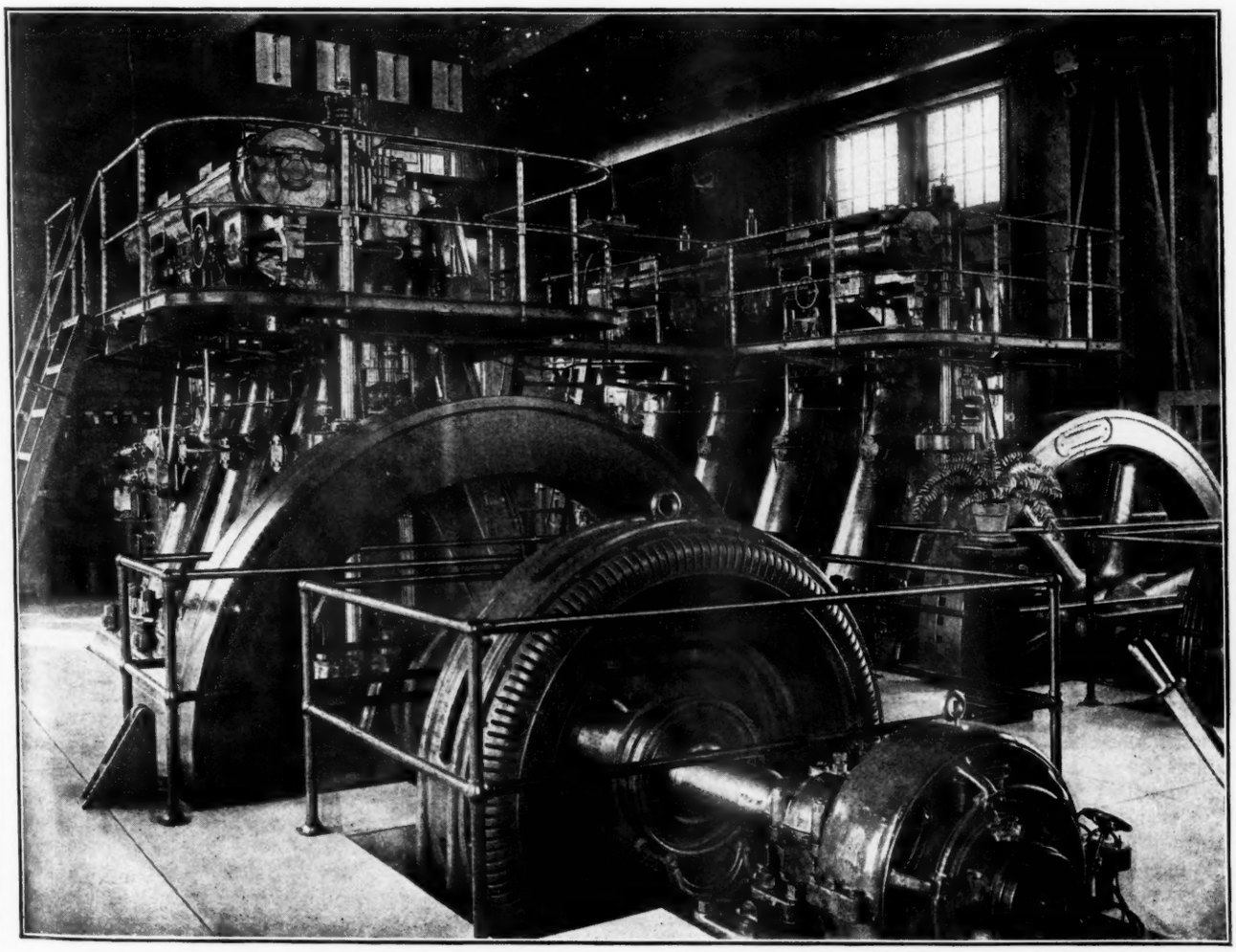


FIG. 1—DIESEL ENGINES IN CROWLEY, LA., MUNICIPAL LIGHT PLANT

read at intervals of ten minutes and the mean revolutions per minute determined. At least one set of indicator diagrams was taken for each run. The diagrams showed a maximum pressure of approximately 500 lb. per sq.in. The indicated horsepower was not computed, as the diagrams were not perfect and were taken as a matter of record as to fuel injection, load balance, etc.

Electrical readings of the test instruments were taken at intervals of five minutes; and the station instruments

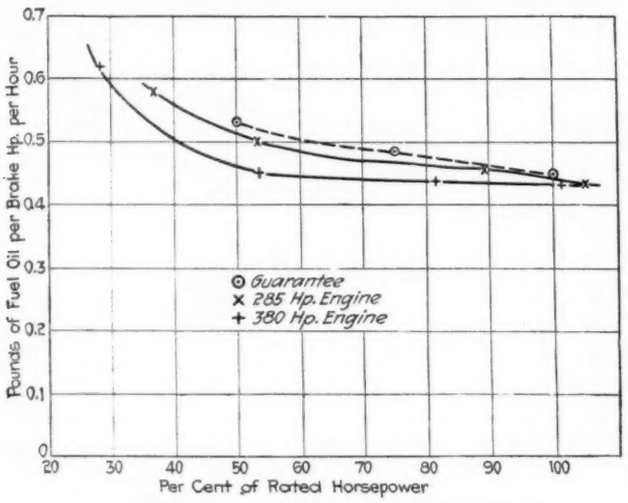


FIG. 2—FUEL TESTS ON FULTON DIESELS

were also read in order to calibrate them with the test instruments.

Standard test instruments from Tulane University were used, there being two wattmeters, a voltmeter and an ammeter. From the readings of the standard instruments the output of the generator under test was deter-

TEST OF FULTON DIESELS, CROWLEY, LA.

Per Cent of Rating	R.P.M.	Gen. Kw.	Gen. Hp.	Exciter Hp.	B. hp.	Fuel Per Hr	Fuel Per B.hp Per Hr
285 H.p. Unit							
100 0	199.9	200.1	291.0	8.5	299.5	128.8	0.430
89.3	201.8	165.7	245.4	8.9	254.3	117.0	0.460
53.6	205.1	97.6	146.1	6.7	152.8	77.4	0.506
36.7	206.2	62.9	98.6	6.0	104.6	60.6	0.579
380-Hp. Unit							
101.3	200.7	261.0	377.0	8.3	385.3	165.5	0.430
81.4	202.4	207.0	302.1	6.9	309.0	135.5	0.439
53.3	205.0	128.5	195.7	6.6	202.3	91.2	0.451
28.3	206.5	65.2	102.8	4.9	107.7	67.5	0.627

mined. Then from the load and power factor the efficiency of the generator became known, as likewise the power given by the engine to the generator. The station instruments were used to determine the load on the exciter, and it was assumed that its efficiency was always 85 per cent. The sum of the horsepowers delivered by the engine to the generator and exciter gave the total b.hp. In some instances a water rheostat was used for the larger loads.

The results are given in the log of the tests. An attempt was made to get full load, three-quarters, one-half and one-quarter loads, but owing to unforeseen changes in load difficult to control, there was some variation from the proposed loads.

The results are plotted graphically, and it will be seen that the fuel guarantees were met in every instance, while in every run but one there was a good safe margin between performance and guarantee. Both engines at a little more than full load used 0.43 lb. of fuel oil per brake horsepower per hour.

The regulation guarantee was met in the 380-hp. unit, which operated at 200.7 r.p.m. at 101.3 per cent load and 206.5 r.p.m. at 28.3 per cent load, a variation of 2.39 per cent.

For the 285-hp. unit at 105 per cent load the r.p.m. were 199.9 while at 36.7 they were 206.2, a variation of slightly more than 3 per cent. There is no trouble in operating these units in parallel.

Effect of Superheat on Steam and Fuel Consumption*

BY R. A. HOLME

EXHAUSTIVE tests on the performance of steam turbines under various degrees of superheat indicate clearly that there is considerable economy due to the use of superheated steam. The theoretical increase in efficiency is very small, probably not over 1 per cent, but a large gain in steam economy results from the elimination of water held in suspension by the expanding steam and the corresponding reduction in friction.

With turbines the steam consumption decreases about 1 per cent for each 10 deg. increase in superheat up to 100 deg. From 100 deg. superheat to 200 deg. the rate of decrease is slightly less, 1 per cent for each 12 deg., while from 200 deg. to 300 deg. the steam consumption decreases 1 per cent for each 14 deg. rise. In other words, the total reductions in steam consumption (over saturated steam) are as follows: 100 deg. superheat, 10 per cent; 200 deg., 18½ per cent; 300 deg., 25 per cent. As these percentages indicate only the savings in steam used by the turbine, there must be deducted the corresponding fall in evaporating performance of the boiler for the respective degrees of superheat.

The following table shows the net gain in coal consumption:

PERCENTAGES OF GAIN AND LOSS

Superheat, Degrees	Economy in Steam Consumption	Fall in Evaporative Performance	Net Gain in Coal Consumption
100	10	3	7
200	18.5	6.5	12
300	25	9	16

The net economy, therefore, in coal consumption by using superheated steam in turbines is 100 deg. superheat, 7 per cent; 200 deg., 12 per cent; 300 deg., 16 per cent.

In the case of reciprocating engines using superheated steam there is again only a slight theoretical increase in efficiency due to the use of superheated steam, roughly about 1 per cent.

To bring data pertaining to the saving in steam consumption in reciprocating engines into line with the figures for turbines, the economy in steam consumption when using superheated steam may be taken as follows: 100 deg. superheat, 12 per cent; 200 deg., 22 per cent; 300 deg., 30 per cent. Or, in other words, the gain due to superheating in reciprocating engines is about one-fifth more than in turbines. Tabulating these figures as before, shows the net gains as follows:

PERCENTAGES OF GAIN AND LOSS

Superheat, Degrees	Economy in Steam Consumption	Fall in Evaporative Performance	Net Gain in Coal Consumption
100	12	3	9
200	22	6.5	15.5
300	30	9	21

*All rights reserved.

Writing Piping Specifications

By HUBERT E. COLLINS*

THE writer has found that many purchasing agents are unable to tell if a specification for material to be purchased for the engineering department is as full as it should be. To cover every detail that is needed when specifying is an art, but to do so saves much time and trouble, and avoids many wordy wars between the engineer and buyer.

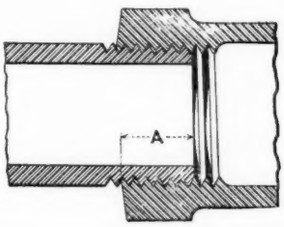
To write a specification requires reference to many trade catalogs and tables as well as a knowledge of what is required. To this end the following ideas are submitted for the guidance, in part at least, of those in the engine room, engineering department or maintenance department who have to do with preparing piping specifications. Credit is given to the Crane Company for the use of several tables.

Wrought Iron and Steel Pipe

When ordering pipe under the heading "Quantity and Size," specify the number of feet of each given size of pipe and note the following points:

Unless otherwise specified the manufacturer will furnish you the required number of feet of pipe in random lengths. In the smaller sizes the random lengths come

Size...Inches	⅛	¼	⅜	½	¾	1	1¼	1½	2	2½	3	3½	4	4½
Price....Each	.05	.05	.05	.05	.05	.06	.07	.08	.10	.15	.20	.25	.35	.45
Size ..Inches	5	6	7	8	9	10	12	14	15	16	18	20	22	24
Price....Each	55	.70	.85	1 00	1 25	1.50	2 50	3 50	3 50	5 00	8 00	10.00	12.50	15 00



Size Inches	Dimension A Inches	Size Inches	Dimension A Inches
¼	¼	3½	1 1/16
⅜	⅜	4	1 1/16
½	½	4½	1 3/16
¾	½	5	1 3/16
1	⅝	6	1¼
1¼	⅝	7	1¼
1½	⅝	8	1 5/16
2	¾	9	1 5/16
2½	¾	10	1½
3	1	12	1 5/8

FIG. 1—THREAD LENGTH AND PRICE LIST FOR THREADING

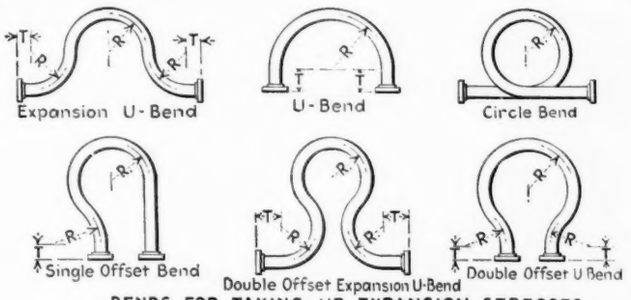
in bundles of suitable size for handling; the total amount of pipe in these bundles, each of which is tagged with the number of feet, will add up slightly less than the amount of the order. The missing amount will be in a random length.

All piping should be ordered shipped with couplings on the threads to protect the latter. Specify protectors for sizes over 3-inch.

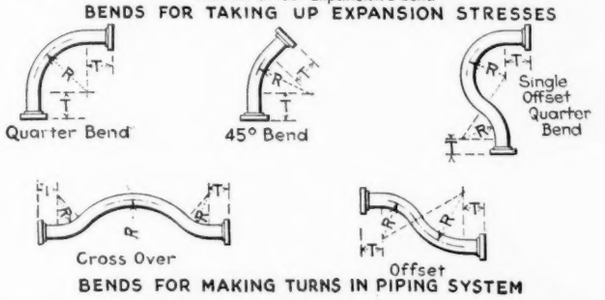
If the pipe is to be cut to order, the exact length must be given in the order or on a sketch attached.

State if cut-to-order lengths are to be end-to-end measurement.

Pipe ordered cut to length should be ordered threaded or not as required.



BENDS FOR TAKING UP EXPANSION STRESSES



BENDS FOR MAKING TURNS IN PIPING SYSTEM

Size of Pipe....Inches		2½	3	3½	4	4½	5	6	7	8	9
R—Minimum Advisable Radius of Bends....Inches		12½	15	17½	20	22½	25	30	35	40	45
Shortest Radius to which Pipe can be bent	*Standard Pipe...Inches	10	12	14	16	18	20	26	30	34	42
	†Extra Strong Pipe....Inches	7	8	10	12	14	15	20	24	28	35
T—Minimum Length of Tangent or Straight Part of Bends	Screwed and Shrink....Inches	4	4	5	5	6	6	7	8	9	11
	Craneweld.....Inches		5	5	5	5	5	6	6	6	6
	Cranelap.....Inches		6	6	6	6	7	7	8	8	9

Size of Pipe....Inches		10	12	14	15	16	18	20	22	24	
R—Minimum Advisable Radius of Bends....Inches		50	60	70	75	80	108	120	132	144	
Shortest Radius to which Pipe can be bent	*Standard Pipe...Inches	45	54	70	75	80	90	104	132	144	
	†Extra Strong Pipe....Inches	40	50	65	70	78	88	104	132	144	
T—Minimum Length of Tangent or Straight Part of Bends	Screwed and Shrink....Inches	12	14	16	16	18	18	18	18	18	
	Craneweld.....Inches	7	7	7	7	8	8	8	9	9	
	Cranelap.....Inches	10	10	14	14	16	18	18	20	20	

FIG. 2—DIMENSIONS OF PIPE BENDS

If threaded cut pipe is wanted, state if it is to come threaded on both ends. This will call for an extra charge for threading and cutting.

All sizes to 12-in. inclusive mean inside diameters. Above 12-in. the sizes given mean outside diameter (O.D.). With O.D. pipe orders the thickness should be specified. O.D. pipe orders will come plain ends unless otherwise specified.

When ordering threads on pipe cut to length, do so according to the dimensions shown in Fig. 1.

When properly cut, the threads should make up in the valves or fittings to the distance given and still show one or two threads outside.

The list prices for cutting these threads extra appear in Fig. 1. At present there is no discount from this list save for 8-in. pipe and larger, for which the discount is from 10 to 25 per cent. These prices include the cutting of the pipe to be threaded.

Brass and copper pipe are made up to thread the same as wrought iron or steel. Either fits iron fittings.

*Consulting Engineer, Utica, N. Y.

Size of Pipe	Minimum Radius Inches		Radius of Bends Inches										
Inches	Standard Pipe	Extra Strong Pipe	20	30	40	50	60	70	80	90	100	110	120
2½	10	7	3/8	7/8	1½	2¼	3¼	4½	5⅞				
3	12	8	3/8	11/16	1 1/16	1⅝	2¾	3⅝	4¾	6			
3½	14	10	7/16	5/8	1	1⅝	2⅜	3¼	4¼	5⅜			
4	16	12	¼	½	11/16	1 1/16	2⅛	2⅞	3¾	4¾	5¾		
4½	18	14	¼	7/16	⅞	1 1/16	1⅞	2½	3⅜	4¼	5¼		
5	20	15		3/8	¾	1 1/16	1¾	2⅜	3	3⅞	4¾	5¾	
6	26	20		3/8	5/8	1	1½	2	2½	3¼	4	4¾	5¾
7	30	24			7/16	⅝	1 1/16	1¾	2¼	2⅞	3½	4¼	5¼
8	34	28			½	¾	1⅛	1½	2	2½	3	3¾	4⅝
10	46	40				5/8	⅞	1¼	1½	2	2½	3	3½
12	54	50				½	¾	1	1⅜	1⅝	2	2½	3
14	70	65						⅞	1⅛	1½	1¾	2⅛	2½

FIG. 3—EXPANSION TAKEN CARE OF BY BENDS

When ordering pipe for bends of various types and uses, the pipe thicknesses should be carefully considered. The thicknesses given in Table I are the lightest the Crane Co. recommends and carry a sufficient safety factor, taking into consideration the scaling of the pipe in the bending fire and thinning of the wall due to stretching during the bending process.

All bends should be ordered with a sketch accompanying the order showing the type of bend desired, together with the principal dimensions. Fig. 2 depicts some of the common types together with the principal dimensions.

Fig. 3 will give an idea of the expansion taken care of by various-sized expansion bends and applies specifically to quarter bends. For U-bends multiply the expansion values given in Fig. 3 by 2. For single-offset bends or expansion U-bends multiply the values by 4. For double-offset bends or circle bends multiply the expansion values by 5.

The expansion shown does not take into consideration springing the bend when bolting into place. When bends are sprung a distance equal to the value given, they will take care of twice that value.

For information on the expansion to be expected in steam piping, Fig. 4 is convenient and is compiled from the results of tests.

EXPANSION OF PIPE
INCREASE IN LENGTH—INCHES PER 100 FEET

Temperature, Degrees F.	Steel	Wrought Iron	Cast Iron	Brass and Copper
0	0	0	0	0
20	.15	.15	.10	.25
40	.30	.30	.25	.45
60	.45	.45	.40	.65
80	.60	.60	.55	.90
100	.75	.80	.70	1.15
20	.90	.95	.85	1.40
40	1.10	1.15	1.00	1.65
60	1.25	1.35	1.15	1.90
80	1.45	1.50	1.30	2.15
200	1.60	1.65	1.50	2.40
20	1.80	1.85	1.65	2.65
40	2.00	2.05	1.80	2.90
60	2.15	2.20	1.95	3.15
80	2.35	2.40	2.15	3.45
300	2.50	2.60	2.35	3.75
20	2.70	2.80	2.50	4.05
40	2.90	3.05	2.70	4.35
60	3.05	3.25	2.90	4.65
80	3.25	3.45	3.10	4.95
400	3.45	3.65	3.30	5.25
20	3.70	3.90	3.50	5.60
40	3.95	4.20	3.75	5.95
60	4.20	4.45	4.00	6.30
80	4.45	4.70	4.25	6.65
500	4.70	4.90	4.45	7.05
20	4.95	5.15	4.70	7.45
40	5.20	5.40	4.95	7.85
60	5.45	5.70	5.20	8.25
80	5.70	6.00	5.45	8.65
600	6.00	6.25	5.70	9.05
20	6.30	6.55	5.95	9.50
40	6.55	6.85	6.25	9.95
60	6.90	7.20	6.55	10.40
80	7.20	7.50	6.85	10.95
700	7.50	7.85	7.15	11.40
20	7.80	8.20	7.45	11.90
40	8.20	8.55	7.80	12.40
60	8.55	8.90	8.15	12.95
80	8.95	9.30	8.50	13.50
800	9.30	9.75	8.90	14.10

FIG. 4—EXPANSION OF PIPE, INCHES PER 100 FEET OF PIPE

TABLE I—THICKNESS OF PIPE FOR VARIOUS BENDS

Radius	Pipe Size	Pipe
	Up to 125 Lb. Working Pressure	
4 to 5 dia.	7-in. and smaller	Extra-strong
	8-in. and smaller	½-in. thick
Over 5 dia.	7-in. and smaller	Full weight
	8-in.	28.85 lb. per ft.
	10-in.	40.48 lb. per ft.
	12-in.	49.56 lb. per ft.
	14-in. to 16, incl.	5/16 in. thick
	18-in. to 22, incl.	⅜ in. thick
	24-in. to 30, incl.	7/16 in. thick
	125 to 250 Lb. Working Pressure	
4 to 6 dia.	7-in. and smaller	Extra-strong
	8-in and larger	½ in. thick
Over 6 dia.	7-in. and smaller	Full weight
	8-in.	28.55 lb. per ft.
	10-in.	40.48 lb. per ft.
	12-in.	49.56 lb. per ft.
	14-in. to 16, incl.	⅜ in. thick
	18-in. to 22, incl.	7/16 in. thick
	24-in. to 30, incl.	½ in. thick
	250 to 350 Lb. Working Pressure	
4 dia. and over	7-in. and smaller	Extra-strong
	8-in. and larger	½ in. thick

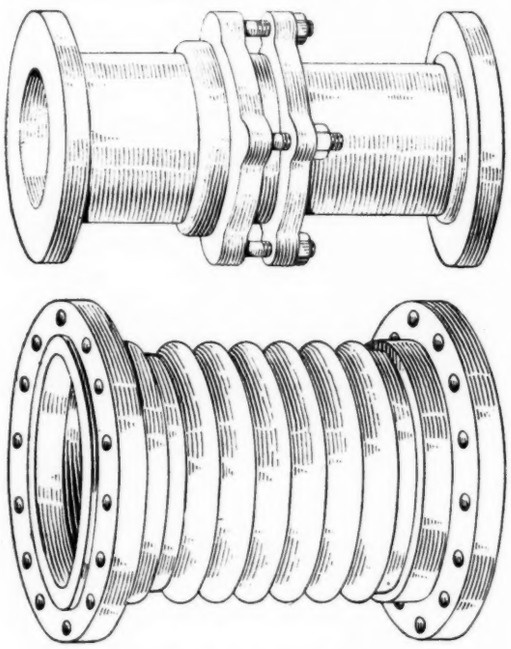


FIG. 5—TYPES OF EXPANSION JOINTS

Either single or double-slip expansion joints or corrugated copper expansion joints may be used in lines when bends and offsets are not practicable. The allowable traverse or movement of these joints determines the number to be installed or the lineal feet of pipe for which each joint may compensate. Joints of the single-slip type are made up to allow a maximum traverse as follows (expressed in inches):

Pipe Diameter .. 2 -2½-3 -3½-4 -4½-5-6-7-8-9-10-12
Traverse..... 2½-2½-2¾-3 -3½-3½-4-5-6-7-7- 7- 8

Joints may be specially made up to allow a traverse of from 6 to 18 in. if desired, although it is customary to limit the traverse of one sleeve to from 3 to 4 in.

Double-slip expansion joints are generally designed to allow a traverse of 4 in. on each sleeve for pipe sizes from 1- to 9-in. 3½ in. on each sleeve pipe for sizes from 10- to 16-in.

Slip joints are usually made with iron bodies and brass sleeves, and must have an adjustable packing gland with follower, as shown in Fig. 5. Joints may be furnished screwed or flanged for standard or extra-heavy service.

Heat loss from piping conveying steam or water should be prevented as far as possible by the use of non-conducting coverings. The loss in uncovered lines and the saving that can be effected by insulated lines justifies a considerable investment in pipe covering.

Diameter			Thickness	Circumference		Transverse Areas			Length of Pipe per Square Foot of		Length of Pipe containing One Cu. Ft.	Weight per Foot of Length	Number of Threads per Inch of Screw	Contents in Gallons per Foot of Length
Nominal Internal	Actual External	Actual Internal		External	Internal	External	Internal	Metal	External Surface	Internal Surface				
Inches	Inches	Inches	Inches	Inches	Inches	Sq. Inch	Sq. Inch	Sq. Inch	Feet	Feet	Feet	Pounds		Gallons
⅛	.405	.269	.068	1.272	.848	.129	.0573	.0717	9.44	14.15	2513.	.244	27	.0006
¼	.54	.364	.088	1.696	1.144	.229	.1041	.1249	7.075	10.49	1383.3	.424	18	.0026
⅜	.675	.493	.091	2.121	1.552	.358	.1917	.1663	5.657	7.73	751.2	.567	18	.0057
½	.84	.622	.109	2.639	1.957	.554	.3048	.2492	4.547	6.13	472.4	.850	14	.0102
¾	1.05	.824	.113	3.299	2.589	.866	.5333	.3327	3.637	4.635	270.	1.130	14	.0230
1	1.315	1.049	.133	4.131	3.292	1.358	.8626	.4954	2.904	3.645	166.9	1.678	11½	.0408
1¼	1.66	1.38	.14	5.215	4.335	2.164	1.496	.668	2.301	2.768	96.25	2.272	11½	.0638
1½	1.9	1.61	.145	5.969	5.061	2.835	2.038	.797	2.01	2.371	70.66	2.717	11½	.0918
2	2.375	2.067	.154	7.461	6.494	4.43	3.356	1.074	1.608	1.848	42.91	3.652	11½	.1632
2½	2.875	2.469	.203	9.032	7.753	6.492	4.784	1.708	1.328	1.547	30.1	5.793	8	.2550
3	3.5	3.068	.216	10.996	9.636	9.621	7.388	2.243	1.091	1.245	19.5	7.575	8	.3673
3½	4.	3.548	.226	12.566	11.146	12.566	9.887	2.679	.955	1.077	14.57	9.109	8	.4998
4	4.5	4.026	.237	14.137	12.648	15.904	12.73	3.174	.849	.949	11.31	10.790	8	.6528
4½	5.	4.506	.247	15.708	14.162	19.635	15.961	3.674	.764	.848	9.02	12.538	8	.8263
5	5.563	5.047	.258	17.477	15.849	24.306	19.99	4.316	.687	.757	7.2	14.617	8	1.020
6	6.625	6.065	.28	20.813	19.054	34.472	28.888	5.584	.577	.63	4.98	18.974	8	1.469
7	7.625	7.023	.301	23.955	22.063	45.664	38.738	6.926	.501	.544	3.72	28.654	8	1.999
8	8.625	7.981	.322	27.096	25.076	58.426	50.04	8.386	.443	.478	2.88	28.554	8	2.611
9	9.625	8.941	.342	30.238	28.076	72.76	62.73	10.03	.397	.427	2.29	33.907	8	3.300
10	10.75	10.020	.365	33.772	31.477	90.763	78.839	11.924	.355	.382	1.82	40.483	8	4.081
11	11.75	11.	.375	37.699	35.343	113.098	99.402	13.696	.318	.339	1.456	45.557	8	.....
12	12.75	12.	.375	40.055	37.7	127.677	113.098	14.579	.299	.319	1.27	49.562	8	.....
13	14.	13.25	.375	43.982	41.626	153.938	137.88	16.051	.273	.288	1.04	54.568	8	.....
14	15.	14.25	.375	47.124	44.768	176.715	159.485	17.23	.255	.268	.903	58.573	8	.....
15	16.	15.25	.375	50.265	47.909	201.062	182.655	18.407	.239	.250	.788	62.579	8	.....
....	18.	17.25	.375	56.549	54.192	254.47	233.706	20.764	.212	.221	.616	70.589	....	.....
....	20.	19.25	.375	62.832	60.476	314.16	291.04	23.12	.191	.198	.495	78.599	....	.....
....	22.	21.25	.375	69.115	66.759	380.134	354.657	25.477	.174	.179	.406	86.609	....	.....
....	24.	23.25	.375	75.398	73.042	452.39	424.558	27.832	.159	.164	.339	94.619	....	.....

FIG. 6—DIMENSIONS OF STANDARD STEEL PIPE

Diameter			Thickness	Nearest Wire Gauge	Circumference		Transverse Areas			Length of Pipe per Square Foot of		Nominal Weight per Foot
Nominal Internal	Actual External	Actual Internal			External	Internal	External	Internal	Metal	External Surface	Internal Surface	
Inches	Inches	Inches	Inches	Number	Inches	Inches	Sq. Inches	Sq. Inches	Sq. Inches	Feet	Feet	Pounds
⅛	.405	.215	.095	12½	1.272	.644	.129	.033	.086	9.433	18.632	.314
¼	.54	.302	.119	11	1.696	.924	.229	.068	.161	7.075	12.986	.535
⅜	.675	.423	126	10½	2.121	1.323	.358	.139	.219	5.657	9.07	.738
½	.84	.546	.147	9	2.639	1.703	.554	.231	.323	4.547	7.046	1.087
¾	1.05	742	154	8½	3.299	2.312	.866	.452	.414	3.637	5.109	1.473
1	1.315	.957	.179	7	4.131	2.988	1.358	.71	.648	2.904	4.016	2.171
1¼	1.66	1.278	.191	6½	5.215	3.996	2.164	1.271	.893	2.301	3.003	2.996
1½	1.9	1.500	.200	6	5.969	4.694	2.835	1.753	1.082	2.01	2.556	3.631
2	2.375	1.939	.218	5	7.461	6.073	4.43	2.935	1.495	1.608	1.975	5.022
2½	2.875	2.323	.276	2	9.032	7.273	6.492	4.209	2.283	1.328	1.649	7.661
3	3.5	2.900	.300	1	10.996	9.085	9.621	6.569	3.052	1.091	1.328	10.252
3½	4.	3.364	.318	0	12.566	10.549	12.566	8.856	3.71	.955	1.137	12.505
4	4.5	3.826	.337	0	14.137	11.995	15.904	11.449	4.455	.849	1.	14.983
5	5.563	4.813	.375	00	17.477	15.120	24.306	18.193	6.12	.687	.793	20.778
6	6.625	5.761	.432	000	20.813	18.064	34.472	25.967	8.505	.577	.664	28.573

FIG. 7—DIMENSIONS OF EXTRA-HEAVY STEEL PIPE

½	.84	.252	.294	1	2.639	.766	.554	.047	.507	4.547	15.667	1.714
¾	1.05	.434	.308	1	3.299	1.326	.866	.139	.727	3.637	9.049	2.44
1	1.315	.599	.358	0	4.131	1.844	1.358	.271	1.087	2.904	6.508	3.659
1¼	1.66	.896	.382	00	5.215	2.78	2.164	.615	1.549	2.304	4.317	5.214
1½	1.9	1.100	.400	000	5.969	3.418	2.835	.93	1.905	2.01	3.511	6.408
2	2.375	1.503	.436	0000	7.461	4.684	4.43	1.744	2.686	1.608	2.561	9.029
2½	2.875	1.771	.552	⅛—	9.032	5.513	6.492	2.419	4.073	1.328	2.176	13.695
3	3.5	2.300	.600	⅜—	10.996	7.175	9.621	4.097	5.524	1.091	1.672	18.583
3½	4.	2.728	.636	⅝+	12.566	8.533	12.566	5.794	6.772	.955	1.406	22.85
4	4.5	3.152	.674	⅛—	14.137	9.852	15.904	7.724	8.18	.849	1.217	27.541
5	5.563	4.063	.75	¾	17.477	12.764	24.306	12.965	11.34	.687	.940	38.552
6	6.625	4.897	.864	⅞	20.813	15.315	34.472	18.666	15.896	.577	.784	53.16

FIG. 8—DIMENSIONS OF DOUBLE-EXTRA-HEAVY STEEL PIPE

BURNING BOILER OIL*

Steam Atomizers

WHEN oil was proposed as a boiler fuel, the plan of forcing the oil through a small orifice by pump pressure to obtain a fine spray was tested out. By reason of the defective design this system was a failure, and the necessity of having some expansive agent to tear the oil into minute drops was generally conceded. Since steam generation was the principal

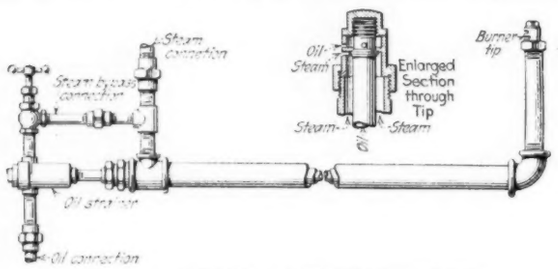


FIG. 1—PEABODY BACK-SHOT BURNER

purpose to which the oil furnace was devoted, steam was the most available atomizing agent and was used in most of the early burners.

One essential feature of a good steam atomizing oil burner is the production of a flame of such shape as to give proper distribution of flame in the furnace and to

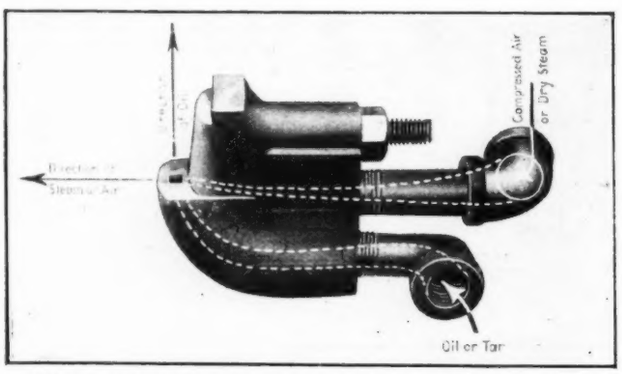


FIG. 2—THE BEST BURNER HAS A SIPHON ACTION

make practicable the reduction of the air supply to a minimum. A second requisite is the necessity of reducing the amount of atomizing steam to a minimum, while a third requisite for efficiency is a burner tip that can be readily adapted to various capacities and shapes

*This is the second article of this series; the first appeared Aug. 7; the third to appear Sept. 4 will discuss mechanical burners.

of flame, so as to fit the burner to the furnace properly.

The other qualities of a good burner are the ones that appeal to the operating man, such as simplicity, ease of cleaning, cheap renewal parts and minimum troubles

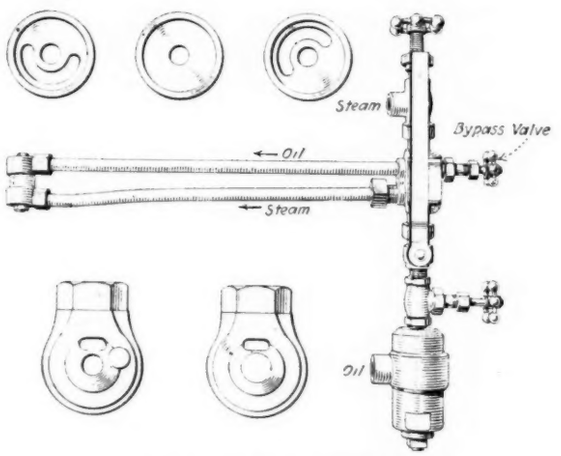


FIG. 3—LIENTZ ATOMIZER

from burner clogging up either through carbonization or through dirt in the fuel.

Steam atomization may be broadly divided into two classes, inside mixers and outside mixers. In the first class the steam and oil mixes in the burner before

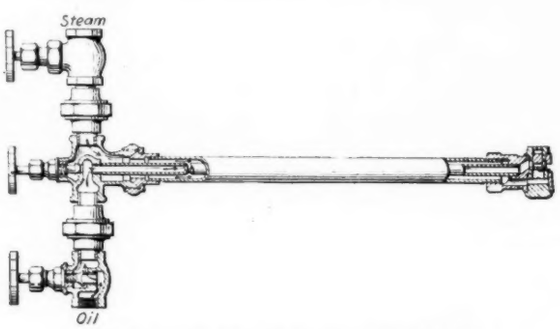


FIG. 4—WITT STEAM-OIL BURNER

issuing in the tip. The second class includes those burners where the air comes in contact with the atomizing oil after issuing from the burner tip. Both these classes might be further subdivided into high- and low-pressure burners.

The Peabody back-shot burner was among the earlier successful atomizers. Since it was developed in connection with inclined-tube water-tube boilers, the nozzle was placed immediately in front of the boiler bridge wall and the oil spray was directed toward the firedoors. In this way the path of the gases before coming into contact with the tubes was increased over the path taken when the burner was directed toward the bridge wall. The atomizer consists of an oil line incased in the steam line, the two lines terminating in the nozzle. This, as will be seen from Fig. 1, contains two slots; the oil issues out of the top slot in a thin stream and meets the steam blowing through the lower slot. The shape of the steam orifice causes the atomized oil and steam spray to take a fantail form of considerable width and comparatively thin. Air for combustion is introduced through checkerwork at the bottom of the furnace. Control is obtained by manipulation of the steam and oil line valves.

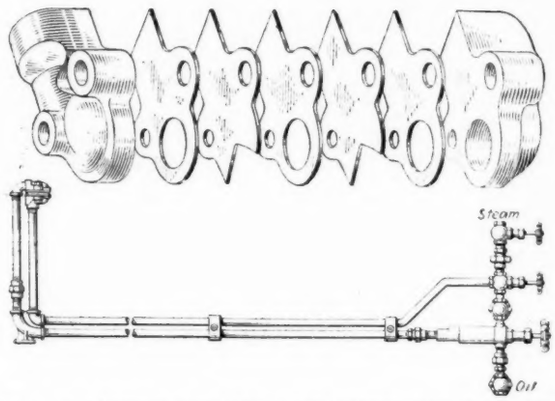


FIG. 5—LOCKETT BURNER ARRANGED FOR BACK-SHOT FIRING

The Best burner, which is of the outside atomizing type, has a two-piece nozzle. The upper lip is movable and in case dirt lodges in the oil or steam line, the lip is moved outward until the orifice is covered. This allows the steam to blow back through the oil line, clearing the orifice.

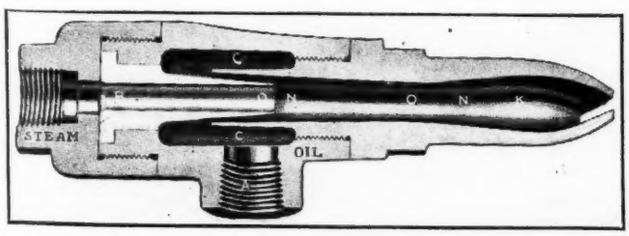


FIG. 6—NATIONAL AIROIL BURNER

The oil enters through the lower pipe connection and issuing out of the orifice, meets the steam spray, which passes out through a slot in the bottom face of the upper casting. The flow of steam across the oil orifice sets up a siphon action tending to increase the oil flow. Consequently, steam pressures as low as 15 lb. and an oil pressure just sufficient to overcome the pipe resistance are ample for atomizing purposes.

Still another outside mixing atomizer is the Lientz. This burner, Fig. 3, consists of oil and steam lines entering the nozzle head. This head consists of two castings separated by three disks, the several parts being held together by a bolt. The oil passes up the lower pipe line into the front casting and then along the upper pipe to the head. Flowing down through the opening in the upper disks, it issues from the slot formed by the center solid disk and the oil disk or ring. The steam passes through the lower pipe and through the hollow steam ring, picking up the oil and delivering it into the furnace in a fantail form. As with most burners regulation is obtained by adjustment of the oil and steam globe valves. In stopping the burner, the bypass is opened, which blows all the oil in the burner out through the orifice. The bypass is also of use in clearing the burner oil tube of dirt, etc.

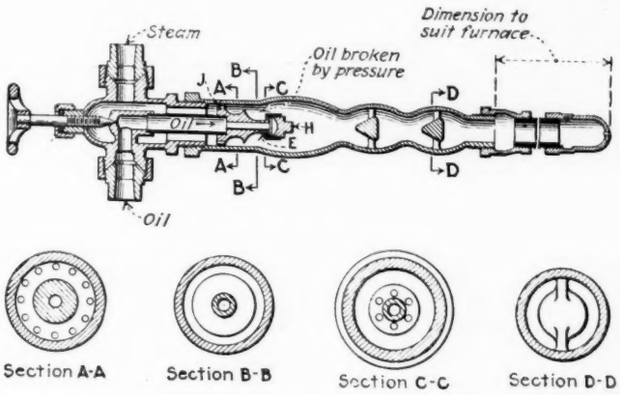


FIG. 7—QUINN STEAM ATOMIZER

A typical outside mixing burner is the Witt, shown in Fig. 4. This burner consists of an oil tube inside the steam tube, together with the atomizer head. The oil issues from the upper slot in the head and meets the steam passing out the lower slot. The resulting spray has a fantail form, although a conical spray is possible by change in the slot shape.

The Lockett outside mixing burner, Fig. 5, was developed in the Southwestern oil fields and is largely used in Southern plants. The burner head contains an oil and a steam cavity, these two passages being separated by a diaphragm. Between this diaphragm and the burner castings are thin steel plates whose functions are to control the capacity of the burner and to regulate the spread of the oil spray. The oil issuing out of the slot formed by the diaphragm and oil plate meets the steam flowing out of the lower orifice. This burner is used both as a front and as a back shot, the difference being merely the piping arrangement.

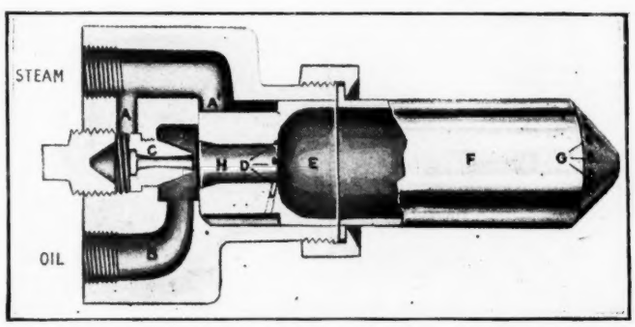


FIG. 8—ENCO BURNER GIVES CONICAL SPRAY

An inside mixing burner is shown in Fig. 6. In this, the National Airoil burner, steam is introduced into the

burner through the passage *B*, which is in principle a divergent nozzle. The steam leaving the end *D* has a high velocity but a low pressure. This draws the oil up through *C*, and the oil steam is churned into a frothy mass in the passages *O* and *K* and issues from the slotted tip in a fantail form. This is essentially a low-pressure burner since the siphon action of the steam flow draws the oil into the burner, and as the oil pressure is low the steam pressure need not be over 15 pounds.

The Quinn inside-mixing burner differs in a number of particulars from the typical burner. As shown in Fig. 7, the oil enters a center tube and issues from a series of holes *E* in the head *H* in a backward direction.

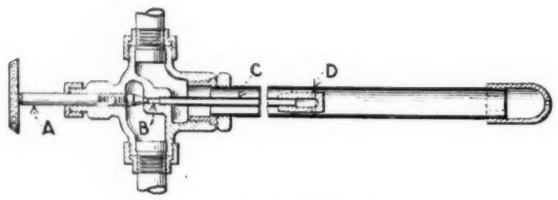


FIG. 9—TATE-JONES BURNER

The steam enters through the top connection and as it passes through the drilled passages *J*, acquires a high velocity, and, when it strikes the oil jetting out from the head *H*, sets up a turbulence that churns the oil and steam into an intimate mass. This mass moves up along the tube toward the orifice, and in so doing the turbulence is increased by the corrugated shape of the tube. The orifice, being a slot, gives a fantail shape to the spray. Since the form of the burner tube is such that it assists the oil and steam mixing, the amount of steam and the steam pressure required are low, from 1 to 1½ per cent of the steam evaporated in the boiler being required for atomizing purposes.

It has been pointed out in *Power*, Aug. 7, that a flat flame restricted the number of burners used per furnace. To enable a hollow-cone spray to be produced from a steam atomizer, the Enco burner, Fig. 8, has been designed. In this burner a stream of steam passing down *A* and through the nozzle tube *C* meets the oil coming up through *B* and a mixture of steam and oil enters the passage *H*. A number of small jets of steam

FIG. 10—FOERST BURNER

issue from the tangential holes *D* and give the mass of oil and steam a whirling motion. This whirling effect continues until the mixture blows out the openings *G* in the form of a series of jets, which unite to form a hollow cone spray. By this design double rows of burners may be placed in a single furnace and so increase the capacity range of the boiler.

The burner manufactured by the Tate-Jones Co. for boiler furnaces appears in Fig. 9. Here oil enters the central tube *C* through the needle valve opening *B* and issues from the head *D* through small passages at right angles to the flow, the steam passing through the outer tube. This mixing of the steam and oil creates a whirling churning which thoroughly atomizes the oil. The

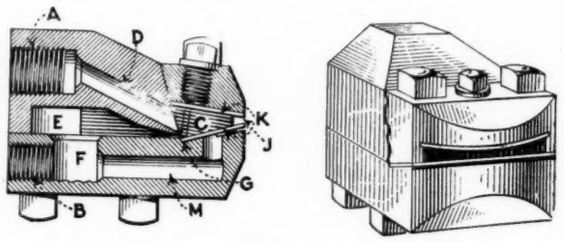


FIG. 11—HAMMEL OIL BURNER

nozzle tip opening is a slot giving the oil spray a fantail form.

The John Foerst & Sons Co. manufactures the steam atomizing burner shown in Fig. 10. This is of the inside mixing type and gives a conical or fantail flame as may be desired.

The Hammel burner, which is of the inside-mixing type, is shown in Fig. 11. Oil enters at *A*, flows through the passage *D* into the mixing chamber *C*. The steam enters at *B*, flows up through *F* and *E* and through three slots at *G* into the mixing chamber. The steam flows across the chamber *C* at a high velocity and thoroughly breaks up the oil, which passes out through the slot orifice at *J*. Steel plates *K K* are inserted in the orifice and may be adjusted to give a spray of the proper thickness. An additional stream of oil passes out below the plate *K* and insures complete atomization of the oil. The oil and steam are controlled by globe

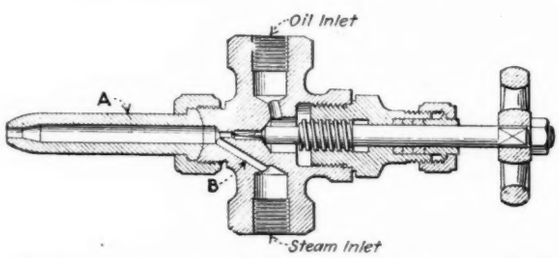


FIG. 12—SCHUTTE & KOERTING STEAM ATOMIZER

valves on the respective pipe lines. This burner is made both for front-shot and back-shot furnaces.

The Schutte & Koerting inside-mixing burner is shown in Fig. 12. As the illustration indicates, the oil flows through a small orifice under needle-valve control into the mixing chamber *A*. The steam jetting up through the passage *B* meets the spray of oil, and a thorough mixing occurs in the chamber *A*. The spray may be either flat or round, as desired, by choosing the proper shape of nozzle opening.

In purchasing oil-burning equipment the amount of steam required for atomization purposes by the burner is of vital importance. Even though the furnace efficiency with a particular burner be 2 or 3 per cent higher than with a second one there might be an actual loss in installing the high-efficiency burner by reason of the large amount of atomizing steam. The range of steam consumption is wide, being from 2 to as high as 10 per cent of the steam generated.

Testing Jet Condensers

By L. LONG
Mechanical and Electrical Engineer, Chattanooga, Tenn.

MANY POWER-HOUSE engineers, particularly in the smaller plants, are inclined to be satisfied with vacuums around 23 or 25 in. The last three inches of vacuum will show more economy on a condensing turbine than any other three inches of the scale, being about 6 per cent less from 28 to 27, 5 from 27 to 26 and 4 from 26 to 25 in. It is necessary, in order to get the best out of the turbine, to keep the vacuum at whatever it has been designed for. Engineers ordinarily agree that there is little to be gained in efficiency by operating a machine designed for 28 in. vacuum at 29.

If the vacuum in a jet condenser is less than 28 in., it is necessary to know the real cause of the trouble, which may be several things in connection with the mechanical elements of the condenser, rather than a vacuum leak. Therefore, do not waste time in taking a candle to look for air leaks, but make preparations to locate the exact causes responsible for these troubles.

For the purpose of reading the vacuum, drill a connection to the exhaust-steam chamber, for the mercury column, and not more than 6 in. from this drill and tap a hole for a ¾-in. pipe fitting, for receiving a thermometer well. Usually, the best place to drill is in the pipe leading to the atmospheric valve about 6 in. from the body of the condenser, just outside of the exhaust steam, as shown in the sketch.

If you have no mercury column, make one by purchasing a 30-in. piece of glass tubing and connect it to the iron piping leading from the condenser, by a short piece of rubber tubing. Thoroughly shellac the glass and iron piping before putting the rubber tubing on it to insure against air-leaks. Insert the free end of the glass tubing in a small bottle of mercury. The height of the mercury in the glass tube may be measured with a rule, or a scale may be made on the frame just behind it. Before the shellac on the tubing dries, tie a cord tightly around the rubber and glass and the rubber and iron to further insure against air leaks.

A valve should be placed on the pipe leading from the condenser to the mercury column as near the condenser as possible. Be sure the bonnet is well packed. With vacuum on the condenser open this valve and the mercury will rise in the column; then close it and if the mercury in the column drops suddenly, you may know that there is a leak in the line of piping leading to the mercury column. All air leaks must be eliminated before the column will be of any service. If an absolute pressure gage can be made as described in *Power* of June 27, 1922, it will be better, as no barometer readings will then be necessary.

Drill holes for ¾-in. pipe fittings in the injection-water pipe about 6 in. from the body of the condenser and, if a rotary air pump is used, in the injection line to the air pump. Screw thermometer wells in all holes after tapping for pipe thread.

Near the thermometer well in the main injection line drill a hole for connecting to another mercury column. In the steam pipe leading to the turbine as near the throttle as possible, place a thermometer well. Connect up a steam gage that has been recently calibrated for reading main pressure, as in the sketch. Inlet steam pressure is read between the throttle valve and nozzle. Pipe plugs are usually provided in the cylinder for such a gage.

For checking cooling water and air tension, the quantities required are given in Table I.

For the purpose of checking, take about five readings, 15 minutes apart. These should be taken with as much load on the turbine as it is possible to obtain, except when testing for leaks in the turbine, which will be described later. Mechanical condition of the condenser refers to that of the air pump, air leaks, mixing of steam with cooling water, etc., and can be readily checked by comparing the absolute pressure maintained within, to the lowest absolute pressure possible to obtain. A jet condenser in good condition is ordinarily guaranteed to hold an absolute pressure of not more than 0.3 in. mercury higher than the lowest absolute pressure possible.

SINCE mechanical conditions in a jet condensing equipment are just as likely as air leaks to cause low vacuum, it may be a waste of time to test for air leaks without first investigating to see where the real trouble lies. Means of locating faults, as here described, may eliminate unnecessary work and delay in restoring normal conditions. Eroded or plugged injection nozzles, high temperature of incoming water, corrections for vacuum and barometer readings and other points that come up in jet condenser testing are described in a direct practical manner.

Steam cannot be condensed at any cooler temperature than that of the cooling water. The temperature of discharged cooling water is taken as the standard temperature for reference, although the average temperature of condensed steam must lie somewhere between this and the temperature of injection water.

Air and Vapor Pressures

If the discharge water is 96 deg. F., the absolute pressure of steam condensed at this temperature is 1.72, expressed as inches of mercury, or in pounds per square inch it is 0.838. If a steam table is convenient, it will show that saturated steam at 96 deg. has this absolute pressure. This is referred to as the "vapor tension," also as "vapor pressure."

In a perfect condenser without air and exhausted to vapor pressure, the exhaust steam would have this absolute pressure or possibly a little lower. The dry air pump and the efficiency of mixing steam and water in the condenser, are not perfect, so that this absolute pressure is not attained when actually condensing steam. If the exhaust-steam absolute pressure is not more than 0.3 in. mercury higher than the vapor pressure corre-

sponding to discharge-water temperature, the condenser is in good condition. This difference, 0.3 in., is termed air tension or air pressure.

Suppose the absolute pressure in the condenser steam chamber is 1.989 in. mercury. This lacks 0.269 in. of 1.72 in., the lowest possible absolute pressure, or vapor tension with 96 deg. discharge water just mentioned. It is well within the limit of 0.3 in. and would indicate a good condenser, although not theoretically perfect. The absolute pressure of exhaust steam referred to as exhaust pressure, is equal to the sum of air and vapor tension; 1.989 in. = 0.269 + 1.720. Too much air tension (more than 0.3 in.) indicates bad mechanical condition.

Low Vacuum Due to Hot Cooling Water

Suppose the condenser were in good mechanical condition but that the discharge water were 126 deg. F. The steam table shows absolute pressure of steam at this temperature to be 2 lb. per sq.in., or 4.07 in. mercury, giving a 26-in. vacuum with a 30-in. barometer, so that even with a theoretically perfect condenser the vacuum would still be too low for good economy. Such a condition is due to insufficient cooling water or the injection water being too warm or to both causes in combination.

TABLE 1—QUANTITIES REQUIRED FOR TESTING JET CONDENSER CONDITION

Load in kw. or hp.
Barometer reading at the level of the condenser, and the hour of the day the readings are taken.
Exhaust steam vacuum indicated by height of mercury column, in inches.
Injection-water temperature.
Discharge-water temperature.
The rise, difference in the injection water and discharge-water temperatures.
Temperature and pressure of steam at throttle.
Temperature of injection water to air pump, if of the LeBlanc type.
Height of mercury in mercury column on injection line to condenser.
Temperature of exhaust steam.
Terminal difference, or that between the temperature of the exhaust steam and temperature of the discharge water.
Superheat, the difference between the temperature of the steam as taken at the throttle, and the temperature corresponding to the gage pressure as shown by the steam table.

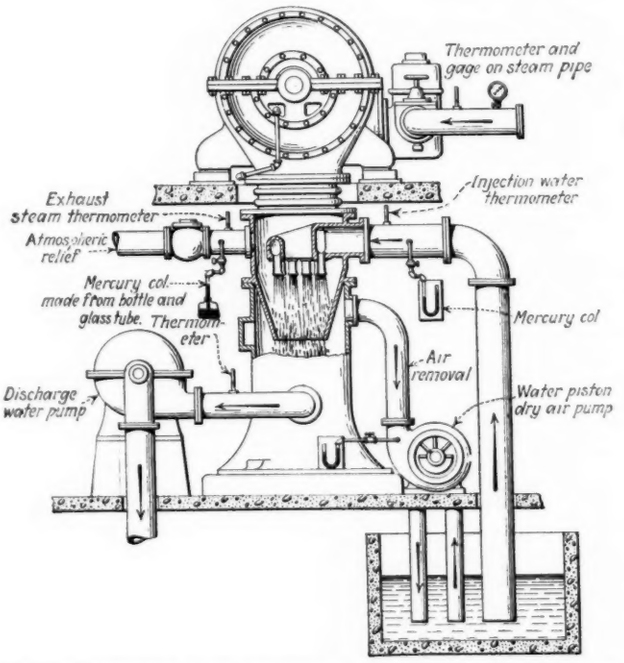
Absolute pressure corresponding to discharge-water temperature is spoken of as vapor pressure or tension, so that this trouble can be described as that due to excessive vapor tension. Condensers are usually provided with cooling water to maintain a vapor tension of 1.7 in. mercury or less. Under favorable conditions such as might occur in winter, a discharge-water temperature of 60 deg. would mean a vapor tension of 0.178 lb. per sq.in., or 0.35 in. mercury. If the discharge water exceeds 96 deg. F., the cooling-water conditions should be checked.

Insufficient water may be due to the strainer being stopped or circulating-pump trouble. It may be advisable to check the pump capacity against guaranteed performance. This may be done by measuring the water discharge by means of a barrel or small tank, after mechanical defects are eliminated.

Checking Mechanical Condition

The exhaust-steam temperature should check with the mercury column; that is, the temperature taken in the well just outside the path of steam from the turbine must show the same vacuum on the steam table that the mercury column shows in inches as measured by the rule.

The temperatures referred to in this article are all based on degrees F.; if readings are taken with a centigrade thermometer, correct by multiplying deg. C. by 1.80 and add 32 deg. Suppose your readings were the following: Vacuum, 27 in.; temperature of discharge water, 96 deg.; exhaust steam temperature, 115 deg. From the steam table you will find that the temperature of the exhaust and that corresponding to the vacuum at the mercury column correspond to each other, therefore you have the correct vacuum. The absolute pressure corresponding to the discharge-water temperature is 1.710 in.; the absolute pressure corresponding to the exhaust-steam temperature is 2.995 in. Air tension = exhaust pressure — vapor tension = 2.995 — 1.710 = 1.285 in. The excess air tension is 1.285 — 0.3 = 0.985 in. mercury, which is much more than the air tension of 0.3 in. allowed. This may be caused by an air leak or by the air pump not being up to capacity. If, on the other hand the discharge-water temperature reads 111 deg., then the vapor tension would be 2.672 in. Take from this the allowable 1.7 in. vapor tension and there is 0.972 in. vapor tension too much. If the pumps are up to capacity, probably some of the spray nozzles are cut or worn.



THERMOMETERS AND MERCURY COLUMNS UTILIZED FOR LOCATING JET CONDENSER TROUBLES

When the injection nozzles are in good condition, the friction head should not be more than 6 in. of vacuum by mercury column through them; that is, there should not be more than 6 in. difference between the mercury column attached to the atmospheric pipe line leading from the body of the condenser and the mercury column attached to the injection-water pipe near the condenser. If there is more than 6 in. drop, look for stopped nozzles; if less, look for the nozzles that have become loose and dropped down in the pump body. Either one of the foregoing will cause a rise in the air tension, because the water does not make proper contact with the steam, therefore fails to cool it sufficiently. Sometimes vanes in the nozzles that give the whirling spray effect have become worn, or in some instances are entirely gone, which allows the water to pass through in a solid stream, thus losing its cooling effect on the steam.

Before the proper reading of the mercury columns

can be obtained, it will be necessary to correct them by the barometer, as all steam tables are based on the assumption that the barometer stands at 30 in. and therefore all steam tables are referred to a 30-in. barometer. The primary object in reading the mercury column is to find the absolute pressure in the condenser. If the barometer is 30 in. and the mercury column is 25 in., then we have 5 in. absolute pressure inside the condenser, roughly 2½ lb. per sq.in. Obviously, if the air pressure is such that the barometer falls below 30 in., then this amount must be added to the mercury-column readings. When a barometer stands at 29.6 in. and a five-inch pressure exists inside the condenser, the mercury column would stand at 24.6 in. Therefore add the amount that the mercury stands in the barometer below 30 in. to the height of the mercury in the testing column; this gives the correct vacuum to use when referring to the steam table. Thus 5 in. + (30 — 29.6) = 5 + 0.4 = 5.4 in., corrected reading.

Likewise, if the barometer stands above 30 in., subtract the amount above 30 in. from the height of the mercury in the mercury column to get the correct vacuum readings. In case the barometer reading must be corrected for altitude also, allow 724 ft. for each drop of one inch in the mercury. This is not absolutely correct, as the farther we get from the earth the more rarefied the air becomes and the greater the distance to be traversed to get the one-inch drop, but this will do for practical purposes. First obtain the barometer readings as shown by the nearest city paper; one of them will be given at one o'clock no doubt. Try to take the condenser reading at or near the time that the observer at the station takes his barometer readings.

Find the altitudinal difference between the mercury column and the barometer and correct as if the barometer were on a level with the mercury column. The mercury column is 500 ft. above the barometer and 500 ÷ 924 will give the fractional inch at which the barometer would stand if it were located at the level of the mercury column; therefore add this fractional inch to the mercury column. If an absolute mercury pressure gage is used, simply assume that the barometer stands at 30 in.; that is, if the mercury gage shows 2 in. absolute pressure inside the condenser, then the vacuum is 28 in. by mercury column.

Air-Pump Pressure Less Than Condenser

Usually, a condenser will get a vacuum around 21 to 25 in. without the assistance of the air pump, provided the water conditions are good and there are no air leaks in it. The air pump then must get a better vacuum than the condenser or else there will be no air to pass from the body of the condenser to the air pump. Fluids and gases pass from one vessel to another by reason of the difference in absolute pressure inside these vessels. A pump creates a pressure, by the movement of the piston, which is less than atmospheric pressure, and therefore water passes from the vessel to the pump. So, for air to pass from the body of the condenser to the air pump, the pressure in the air pump must be less than the pressure in the condenser, and the air pump must produce a better vacuum than you are expecting in the condenser.

If you wish 28 in. then the air pump must obtain a vacuum better than 28 in. or there will be no air to pass from the condenser body to the air pump, which must also have enough capacity to remove the air entering with the steam. About 5 per cent of the total mixture entering the condenser in the turbine exhaust is air; if 20,000 lb. of steam is entering the condenser per hour, then the pump will have to remove about 1,000 lb. of air per hour. It must do this and still keep up a vacuum of more than the 28 in. required in the body of the condenser.

To test the air pump for vacuum, blank it off from the condenser by placing a blank gasket in the air intake pipe; reinforce the gasket with a steel flange. Drill a hole in the air pipe next to the pump, connect it to the mercury column, and start the air pump. A LeBlanc or other similar type of pump will get a vacuum corresponding to the injection-water temperature when in good condition. If the injection water is 70 deg., the pump should get a vacuum of 29.259 in. referred to a 30-in. barometer. A reciprocating pump should get a vacuum of about one inch more than that expected in the condenser body.

To find if the pump is up to capacity, test it with an orifice. The pump manufacturer will give the size of the orifice for the different pumps and the amount of drop in vacuum to be expected from the insertion of the orifice. If no standard orifice is obtainable, make one by filling a thermometer well with babbitt and drilling a hole the size required. The bit should not wabble, or it will make the orifice too large.

Testing for Air Leaks

Fill the condenser with water up to the turbine glands, and if there are any leaks the water will find them. It will do no good to run the turbine non-condensing, as the expansion from heat will close up all air leaks so that no steam will show. Only once have I received any benefit from testing with a candle; this was forced on me for the reason the chief engineer would not let me close the turbine down and fill it with water. The leak was all in one place, which was large enough to find with a candle.

If a reading is taken at full load, part load and light load, and there is an increase of air tension as the load decreases, you will find that the turbine is leaking along the central flange. At very light load the vacuum will extend almost as far as the intermediate blades, and if there are leaks in the casing at or near the exhaust chamber they will admit air under light load.

To find the amount of steam entering the condenser through the turbine, determine the area of the nozzles, and inlet pressure, and use one of the three formulas here given, the first two of which may be found in any handbook on engineering.

Napier's formula, steam in pounds per second equals area in square inches of the throat × pressure absolute ÷ 70. Expressed in symbols $S = \frac{AP}{70}$ where P is the absolute pressure effective at the throat. Grashof's formula, same notation, is $S = 0.0165\ AP^{0.97}$. Westinghouse steam chart calculated from Peabody's may also be used. All three of these methods will give a slight difference in steam flow through the same area of nozzles, but either one is an accurate way of finding the steam flow. Grashof's gives the highest, Westinghouse next and Napier's the lowest amount. For this reason I prefer the Westinghouse chart. There is not enough difference between them to amount to anything. If the area of the nozzles cannot be obtained, draw a steam-flow curve from contract guarantees and find the amount of steam at the kilowatt load for the reading

taken. A steam-flow meter may be used if it has been recently calibrated.

If the discharge water pumps are not up to capacity, see if they are up to speed. When a centrifugal pump is used, take the cap off and examine the seal rings. These may be rubbed by allowing the bearings to wear and produce contact between them. If the pump has been run at any time without water, they may heat and burn out and the water will then circulate from the discharge chamber to the intake chamber, reducing the capacity of the pump. Keep a calibrated low-pressure gage or preferably mercury U-tube, if the pressure is low enough on the discharge line. See to it that the pump is not discharging against a head higher than designed for. Gage pressure times 2.31 will give the total head the pump is discharging against, including pipe friction. Examine the runners and see if the blades are worn.

If the discharge line has a swing check valve in it, examine to see if the valve is broken off and lodged in the discharge line. Either locate the gages on the water lines or correct for the height of the water column in the pipe leading to the gage. See if the suction head is greater than the contract calls for. When it is discharging into a spray pond through nozzles, examine the nozzles; they may be filled with dirt, lime or other sediment. If the discharge pumps are located in the bottom of the condenser, be careful to keep the glands properly packed and sealed with water, or some air may leak in by the shaft. Test the atmospheric valve for leaks by observing if sealing water leaks through excessively and also by listening for air suction.

The American Suspended Arch

A suspended sprung arch designed to eliminate the lateral compression of the sprung arch and to reduce spalling from unrelieved pressure, was described in the Oct. 19, 1920, issue of *Power*.

Since that time the patents and manufacturing rights of this arch have been acquired by the American Arch Co., Inc., 17 East 42nd St., New York City. Several substantial improvements have been made in the design, and it is now known as the "American Suspended Arch,"

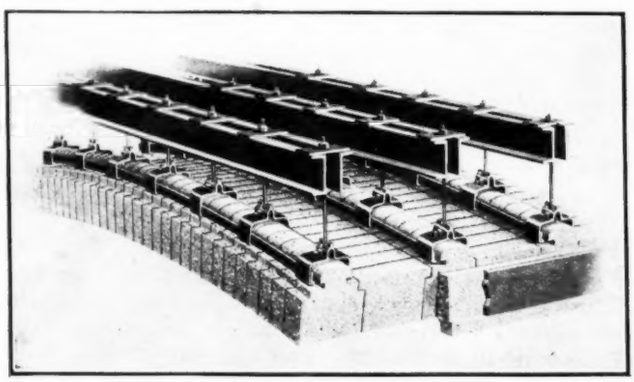


FIG. 1—SHOWS METHOD OF SUSPENDING THE MAIN ARCH

and is being extensively manufactured for use in boiler furnaces.

Probably one of the most important improvements is in the method of suspending the arch. This is clearly shown in Figs. 1 and 2. A substantial extended head is formed on each brick in alternate rows across the furnace. Heavy-gage sectional angle runners *A* approximately 18 in. in length engage the heads of the brick. Each pair of angles supports six bricks and is in turn supported at both ends by pressed steel clips *B* hung from I-beams above the arch. This gives a very sturdy construction with all the suspension means visible from the top of the arch. The use of interlocking filler bricks between and supported by the suspended rows has been retained. The small ignition arch shown at the front and the fantail section at the

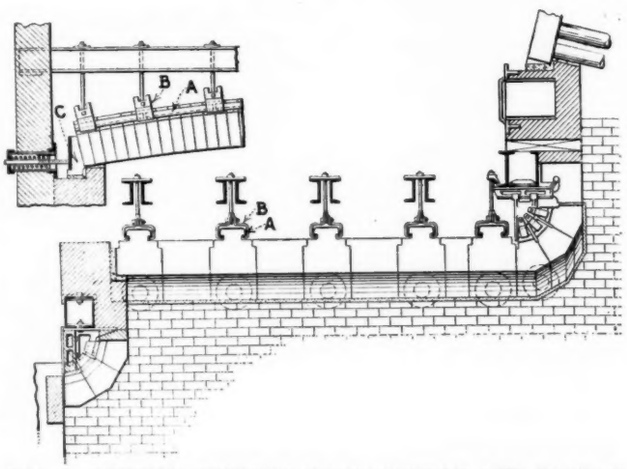


FIG. 2—SECTION SHOWING HOW IGNITION AND FANTAIL ARCH ARE SUSPENDED, ALSO METHOD OF PROVIDING SLIGHT LATERAL PRESSURE

rear below the boiler header are supported in a manner similar to the main body of the arch. Any section of the arch can be readily replaced without disturbing a large number of bricks.

The method of providing a slight pressure at each side of the arch has been greatly improved by placing a series of compression springs in the side walls, as shown in the upper left-hand view of Fig. 2. These springs exert a steady pressure against the skew brick *C* and tend to maintain a tight brick-to-brick joint at all times, taking care of contraction or shrinkage and at the same time allowing for expansion without binding on the side walls. As the entire weight of the arch is carried vertically, the pressure required is nominal—about 2½ lb. per sq.in. of skew area.

The bricks in the main arch are slightly tapered, and there is sufficient pressure to hold them in place should they become checked or cracked from sudden variations in temperature.

The arch is laid to a large standard radius with only a nominal rise. When required to meet special conditions, as under low-set boilers or to accommodate vertical powdered-coal or refuse burners, this arch is also installed flat.

Scale formation at the River Rouge plant of the Ford Co., and the accumulation of foreign matter in the boilers is reduced to a minimum by distillation of the make-up water and utilizing practically 80 per cent of the condensation from steam generated as feed water. Approximately 2,600,000 lb. or 1,300 tons of make-up water is required daily. The total steam production is approximately 13,000,000 lb. per 24 hours, during the winter months.

A High Pressure Mercury Reducing Valve

By J. W. Mackenzie*

To overcome the wide pressure fluctuations of mechanical valves from boiler-feed water pumps, the Consumers Power Company designed a mercury-controlled valve which maintains practically a uniform pressure on the boiler-feed line. During the reconstruction of one of the power plants it was found desirable to use motor-driven centrifugal feed pumps. The pumps chosen for this purpose were designed to give as flat a head curve as possible throughout the full range of capacity, and at the same time show a good efficiency over the entire normal operating range of the pump.

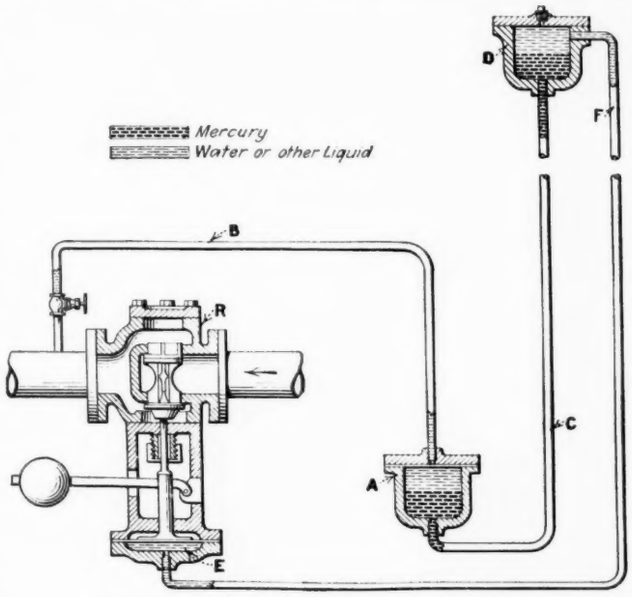
The designers had foreseen that notwithstanding this flat head characteristic, there would be a variation of about 15 per cent between the no-load and the full-load discharge pressure. This condition was due in part to the characteristics of the pump and also in part to the slightly lower motor speeds under full load which might be still further reduced at times by low frequency of the electric service. To protect against this condition and also to insure full feed-line pressure at full capacity when the pumps had become worn, the pumps were built to give a discharge pressure at light loads of about 340 lb. per sq.in. and approximately 310 lb. at full load. A regulating valve was installed in the pump discharge line to reduce this varying pressure to a uniform positive pressure of approximately 285 lb. on the main feed line.

It was decided to maintain a constant feed-line pressure rather than a constant differential pressure between the feed line and the steam header. This was due to the fact that at high ratings when the maximum differential was needed, there was an appreciable pressure drop between the boiler and the header which was caused by the resistance of the superheaters, valves and piping, resulting in a reduced differential pressure. The installation was made according to this arrangement, but difficulty was experienced in maintaining a reasonably constant pressure on the feed line. The valve-control mechanism, in order to withstand the heavy forces resulting from the high pressure, had to be made so heavy that the resultant friction and inertia required a considerable change in pressure to operate the valve. This caused fluctuations in the pressure of between 30 and 40 pounds.

To overcome this difficulty, a new reducing valve was developed and has given satisfactory regulation for more than a year. The equipment consists of an ordinary diaphragm-controlled reducing valve of the balanced type as shown at *R* in the illustration. A small pipe *B* connects the discharge of the valve with the top of a small reservoir *A*. A quarter-inch pipe *C* joins the bottom of reservoir *A* with the bottom of reservoir *D* located at a higher elevation. The water of the upper part of reservoir *D* runs through a small pipe *F* to the diaphragm chamber *E* of the reducing valve.

To put the system in operation, the tops of the reservoirs are removed and a quantity of mercury, a little more than enough to fill the pipe *C* and allow for the diaphragm displacement of the valve, is placed in reservoir *A*, the reservoir is then filled with water and the top replaced. Water pressure is then admitted through pipe *B* until a small amount of mercury is forced up into reservoir *D*. This reservoir, the pipe *F* and the diaphragm chamber *E* are filled with water, and the valve is then held in the open position until the top of reservoir *D* is secured in place. The valve is now ready for operation, since it is only necessary to open the valve in the pipe *B* and adjust the weight on the lever of the valve to obtain the desired pressure.

The mercury is used to balance the heavy water pressure. In this installation, where the pressure desired was 285 lb. per sq.in. the upper reservoir was placed 46 ft. above the lower reservoir, so that about 270 lb. of the water pressure was balanced by the mercury column, leaving approximately 15 lb. pressure in the upper reservoir. The pressure effective on the valve diaphragm is therefore 15 lb. added to the head of water in pipe *F*, or a total pressure at the diaphragm of about 35 lb. This requires only a light diaphragm. An increase of pressure on the outlet of the valve forces



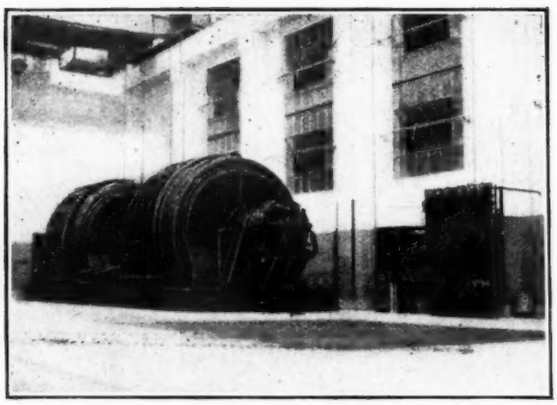
ARRANGEMENT OF REDUCING VALVE AND MERCURY RESERVOIRS

a small quantity of mercury from the lower reservoir up into the upper reservoir, from which the displaced water is forced into the diaphragm chamber *E* through pipe *F*, thus closing the valve; while a reduction in pressure will cause the mercury to flow from the upper reservoir into the lower reservoir, thereupon opening the valve. It will be seen that while the diaphragm chamber is operating under a pressure much less than the pressure in the feed line, any change in pressure in the feed line is transmitted undiminished to the diaphragm.

The reservoirs *A* and *D* are made with sufficient capacity to hold all the mercury in use in the system so that in case the diaphragm should fail, the mercury will be forced into the upper reservoir, allowing the water to bubble through it and escape without forcing the mercury over into the diaphragm chamber or out of the system.

In case more pumps are desired to be operated on the same line, it is only necessary to install a reducing valve with each pump, connect all the diaphragm chambers to pipe *F* and add sufficient mercury to the system to take care of the additional diaphragm displacements. One set of reservoirs is therefore sufficient for any number of valves.

*Operating Department Consumers Power Co., Jackson, Mich.



Aligning a Frequency Changer Set

AFTER grouting and assembling a surveying instrument was utilized for checking approximately the leveling of each bearing. The coupling flanges between the motor and the generator were then disconnected and measured for central and parallel condition. Shims were adjusted under the bearing supports in order to bring these flanges in correct alignment.

THE 15,000-kva. synchronous motor-generator set illustrated at the top of the page changes 50-cycle energy into that of 60 cycles, at 15,000 volts. The outline in Fig. I shows the rigid coupling between the motor and generator. An exciter is also mounted at each end of the set. This operates at 600 r.p.m. and totals approximately 200 tons, the heaviest piece, which is the generator rotor weighing 35 tons.

The bedplate was carefully leveled and grouted in the usual manner after which the machines were assembled. In order to check the alignment roughly, a surveying instrument was used to measure the height of each journal above a certain datum or specified elevation. After correcting for the diameters of the shafts, the center of each journal was found to be practically at the same level.

The shaft deflection at each bearing was measured by means of a sensitive machinist's level 18 in. long. Shims were inserted at the lower end between the journal and the level until this was raised to a level position. An exaggerated plan of shaft deflection is shown at the bottom of Fig. I. The heavy weight of the rotating element in each case tended to deflect the shaft. The deflection at the generator end bearing was found to be 0.014 in. with the exciter end high. The deflection at the generator coupling bearing was 0.007 in. with the generator end of the journal low. This latter deflection represents that required for the entire 18 in. length of level which corresponds to a deflection of 0.0046 in. per ft. Deflections of motor bearings 2-1 and 2-2 were 0.007 and 0.013 in respectively. After making corrections for alignment the shaft deflections were again measured, as will be described later. These deflections indicate only the general conditions of the rotor and are not vitally concerned with the alignment itself.

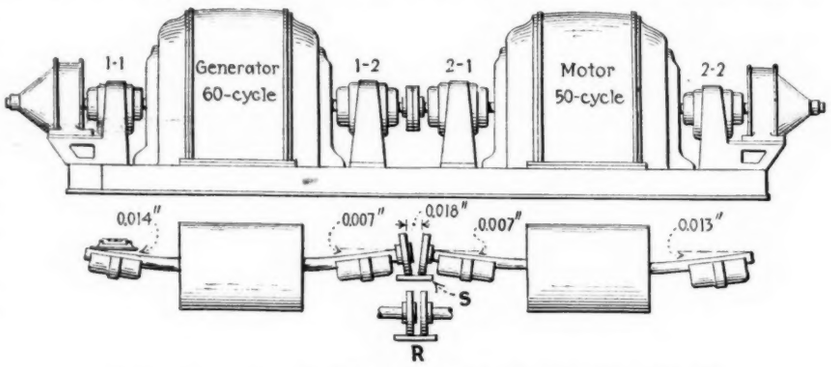


FIG. 1—THE SET WITH SHAFT AND COUPLING DEFLECTIONS SHOWN AS EXAGGERATED

The object of aligning is to place the coupling flanges central with respect to each other and parallel in regard to the flat radial surfaces. The bolts were removed and the two flanges separated $\frac{3}{8}$ in. In order to check the centricity or central condition, a straight-edge is placed across the coupling flanges at the top, the bottom and each side. If the flanges are both parallel and central, the straight-edge will bear evenly throughout its length, as at *R*, Fig. 1. If the flanges are displaced centrally, the straight-edge, when contacting at the high flange, will not bear evenly on the low side of the flange, or in other words, that which is deflected toward the center. Since these flanges were out of parallel, it was possible to make only an approximate check, as at *S*. The flanges appeared to be central.

In order to test the parallel condition, measurements were made between the two flanges with thickness gages, at the top and bottom and both sides. A solid piece was used to fill in most of the $\frac{3}{8}$-in. gap between the flanges, so that only a comparatively small amount of thickness gages were necessary. It will be seen that at *A*, Fig. 2, the flanges separated 0.018 in. at the top and at the bottom 0, while at each side they opened 0.009 in. The flanges were therefore parallel with respect to each other, from one side to the other, but opened somewhat from the bottom to the top. The fact that the opening at the top was 0.018 in. and the opening at each side was 0.009 indicated also that the flanges were machined true, since the sum of the horizontal measurements equals the sum of the vertical.

The next step in alignment is to bring the flanges parallel with each other and then again check the central condition. Liners or shims amounting to 0.055 in. were added under the pedestal block, at bearings 1-1 and 2-2, Fig. 1. This is for the purpose of tipping the exciter ends of both the shafts upward, so that the coupling flanges will then close in at the top and become more nearly parallel. By carefully figuring the proper amount of shims, the flanges in this instance were brought satisfactorily parallel by this one adjustment. Applying a straight-edge at top, bottom and both sides showed the flanges to be satisfactorily central.

In order to check thoroughly the parallel condition of the flanges, a series of measurements were taken as indicated at *B*, *C*, *D* and *E*, Fig. 2. The bearing pedestals were first inspected to make sure that the bolts were

carefully drawn down in place, so that there would be no shifting to disturb the shaft position. The measurement at *B* indicates the flanges to be perfectly parallel, and instead of getting zero measurements all the way around, the top and bottom measurements were 0.0015 more than zero. Both sets of measurements are practically equal. A small error of 0.001 in., and possibly 0.002 in., is usually considered negligible when aligning machines in a case of this kind, although it is desirable to avoid this much. This is especially negligible when the two sets of measurements check against each other and do not indicate an out-of-parallel condition.

Several checks were now taken, as indicated at *C*, *D* and *E*, Fig. 2. Before taking each measurement, the rotor of the motor was given one-quarter turn, which is through a right angle, always in the same direction, while the generator shaft was maintained stationary. This operation is necessary in order to check whether there are any springs or other unevenness in the coupling flanges or shafts. It might happen that the coupling flanges would be parallel while both shafts were sprung in such a manner as to still keep the flanges parallel when measuring at *B*, Fig. 2. In other words, if one shaft were sprung up and the other down, the flanges might still measure as being parallel from top to bottom. In order to check this, one of the coupling flanges is turned through 180 deg., or in other words, through one-half turn. Any spring that exists will then show its error by twice the amount of flange inclination. The readings at *C*, *D* and *E* are found to check parallel within the allotted limits of error. The alignment was therefore considered O.K. with no appreciable shaft springs.

The shaft deflections were then measured at each bearing, with a level, as before described. In this case the bearing at 1-1, Fig. 1, required 0.017 in. in order to bring the bubble to a level position; the bearings 1-2 and 2-1 both showed level at the journal. The bearing 2-2 required 0.0175 in. in order to show level. The top, bottom and side measurements, with a straight-edge, indicated the coupling flanges as being satisfactorily central.

In order to align the rotating with the stationary element of the generator the air gap of $\frac{3}{8}$ in. was adjusted evenly. In order to do this, 0.114-in. liners were necessary under the generator feet. The air gap of the motor was adjusted at $\frac{3}{4}$ in. evenly all around by adding liners of 0.125 in. under the feet.

The exciter on the 60-cycle generator was aligned correctly by adding liners of 0.08 in. between both outboard feet and the pedestal and liners of 0.071 in. between the inboard feet and the pedestal. The exciter of the 50-cycle motor required liners ot 0.0855 in. at the outboard feet and liners of 0.0795 in. on the inboard feet.

The exciter couplings were loosened up and the separation checked as shown at *F*, *G*, *H* and *I*, Fig. 2, for the 60-cycle generator exciter. After each measurement the revolving element was given one-quarter turn. It will be seen that these show the coupling to be very accurately parallel. The flanges were held central by the plug and recess centering fit. The measurements of the 60-cycle motor exciter are indicated in *J*, *K*, *L* and *M*.

After the couplings were properly bolted up, the machine was tested for resistance of the insulation and placed on the line with satisfactory operation. It is notable that in starting this set, oil is supplied temporarily under high pressure, at each bearing. There are two oil grooves, one on each side of the bottom of the liners. Oil under pressure tends to lift the shaft off the journal. Under such conditions the entire rotor can be turned over by one man pulling on the 24-in. coupling. Ease of turning indicates a high degree of accuracy in alignment.

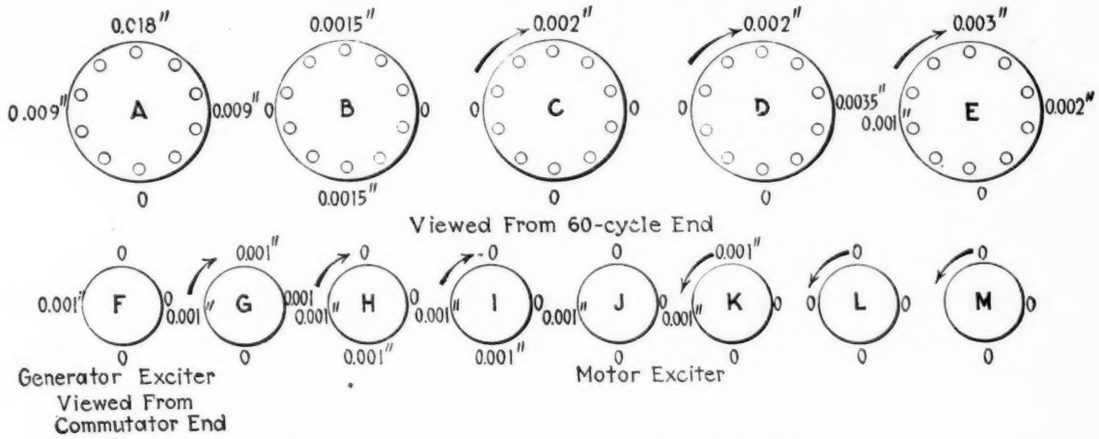


FIG. 2—MEASUREMENTS ON THREE COUPLINGS INDICATE ABSENCE OF SPRINGS OR UNEVENNESS

Heat Transfer in Internal-Combustion Engines

As a result of tests carried on with bombs, Dr. W. Nusselt, in an address read before the *Verein Deutsche Ingenieure*, Berlin, June 29,1923, stated that heat passes into the cold wall of a bomb partly by conduction and partly by radiation. In order to separate the whole transmission of heat into both elements, use was made of bombs blackened on the inner surface and of ones gilded within, the difference of delivery in both cases being proportional to the radiation of heat of the hot gases of combustion.

It has been shown that the radiation of heat from hot gases follows the Stefan-Boltzmann law of radiation—that is, it increases with the fourth power of the absolute temperature—but on the other hand it has also been shown that the amount of radiation by hot gases is only about one-fifteenth of the radiation by a black body of the same temperature. The heat delivered by conduction depends on the temperature and on the pressure.

By making use of the gas-engine tests of Clerk, the influence of the piston speed on the heat given up was taken into consideration. Applying the new formula to the delivery of heat in the combustion engine, it will be seen that radiation of gas does not play the part formerly ascribed to it and that it suffers strong alteration during the combustion period. The assumed dependence of the coefficient of heat transition on the mean piston speed is certainly only an approximation.

Suggestion Systems in Power Plants

By C. M. King

No power plant is so perfectly equipped and operated that valuable suggestions will not be offered by the employees if suitable encouragement or reward is given, and it has been proved in at least one hydro-electric plant that the employees and the management will be mutually benefited by the institution of a suggestion department. Some of the reasons, from the viewpoint of the employee, for the desirability of such a department are: Opportunity is offered him to present suggestions to the management in a business-like, dignified manner; he is encouraged to think of or beyond his work; an excellent opportunity is offered to demonstrate his fitness for promotion; there can be no reason for him to believe that the management is not receptive to suggestions, or to feel that if a suggestion is made to his immediate superior, it may be passed on without his receiving due credit. He also has the personal satisfaction of knowing that, regardless of the apparent importance of his work, his suggestions will be afforded exactly as much consideration as those of the highest executive.

If the suggestion department is earnestly supported by the management, a considerable number of suggestions will be regularly contributed, many of which will be of sufficient value to warrant immediate adoption, and some may start investigations that will result in findings of an important nature or disclose conditions not generally known to exist. Besides indicating, to a great extent, the interest shown in his work, suggestions make it possible to gage an employee's capacity for increased responsibility, and affords the management the opportunity to share the experiences and thoughts of a large number of employees who would otherwise rarely be heard from.

In a large hydro-electric plant forming part of a system on which a suggestion department was instituted several years ago, many visitors commented favorably on the number of original or improved devices and practices in evidence, most of which had their inception in suggestions offered by employees. The suggestion department consists essentially of a small group of men, known as the suggestion committee, and appointed by the general superintendent. In order that it shall be representative of the various departments and to insure correct grading of the suggestions, such men as the station superintendent, transmission superintendent, chief operator and foreman of construction and maintenance constitute the suggestion committee. To prevent any possible abuse of the system, prizes are not regularly awarded to members of the committee or to men whose positions rank higher than those of chief operator or foreman.

Suggestions are written on plain paper, dated and signed, and are inclosed, with any required drawings or sketches, in suggestion envelopes, which are deposited in a suggestion box in the station. The envelopes are removed by a designated member of the committee, opened and, after being typewritten, one copy, together with a photostat or blueprint of each sketch, is delivered to each member of the committee. On the suggestion form there is space for comment or criticism and for the expression of an opinion as to the present or future value of the suggestion.

The committee grades the suggestions quarterly and agrees on the award of the prizes, three of which, the first, second, and third, amount to $10, $7 and $5 respectively, and several prizes of $2 each are available for distribution. The names of the winners are posted on the bulletin boards, and the checks delivered by mail.

After the prizes are awarded, the suggestions are further studied and classified as to their importance, cost and order of adoption, and when possible, an estimate is made of the loss or profit likely to result from the adoption of each. A record of the employee's suggestion rating, based on the value and number of suggestions submitted by him, which is kept constantly up-to-date, serves as a valuable index of his ability and of the interest shown in his work. An unsigned copy of each suggestion, together with any criticism or comment, is placed in a suitable binder in the stations for the perusal of any one interested.

Payment for Suggestions

On some systems an effort is made to induce employees to contribute suggestions without hope of remuneration, but since the management derives benefit from the suggestions, it seems proper to pay for those of value. If, instead of paying fixed sums, the payment is made a definite percentage of the value of the suggestion, it is probable that the maximum number of valuable suggestions would be submitted. It would undoubtedly be difficult to determine the value of certain useful suggestions in dollars and cents, but by establishing a minimum sum as the lowest award a satisfactory scheme could be worked out. The principal advantage of paying a definite percentage of the saving that results lies in the incentive thereby offered the employees to study important problems.

It would seem only fair to share, say 50 per cent of the net saving expected to accrue during the first two-year period following the adoption of the suggestion, with the employee. As an example, by putting into effect a suggestion requiring a change in electrical resistances, a net saving of approximately $400 resulted during a two-year period, due to decreased power consumption by certain auxiliaries in a power plant, and this sum might properly have been shared equally with the employee whose suggestion made the saving possible.

Strange as it may seem, it has been the writer's experience that many men directly in charge of hydro-electric plants receive suggestions with poor grace and, probably due to lack of enthusiasm or interest, actually discourage or block the adoption of the suggestion until possibly an interruption of service, breakdown of equipment or a personal accident clearly demonstrates their value. For this reason satisfactory results cannot be expected from the institution of a suggestion department unless the highest executives earnestly support it and assure themselves that every employee understands that suggestions are desired and welcomed and that valuable ones will be put into effect and suitable compensation given.

EDITORIALS

F. R. LOW, EDITOR

America Shoulders the Scientific Burden

SCIENCE knows no national boundaries. Scientific workers in all countries have for centuries made the results of their researches mutually available. The fruit of this policy may be seen in the world's accelerated advance in science and engineering during the last hundred years. In this each nation followed its own particular bent, availing itself of the common fund of information and experience to which all had contributed. Compared with England, Germany and France, America took a minor, though important, part in the fundamental scientific work. Europe loved science for its own sake as well as for its practical uses, but the main currents of American genius and endeavor were turned to the practical application and development of the findings of European scientists. The practical results have surpassed, at least in quantity, anything that Europe can exhibit, but full credit must go to those scientists who laid the foundations. Faraday would hardly be called a practical man, yet every superpower station is a monument to the memory of this physicist who "played" with wires and magnets.

Without losing her grasp on the purely practical, America has in recent years increased by leaps and bounds her contributions to scientific research. The ever-expanding work of the universities is now rivaled and often surpassed by that of great research laboratories supported by the government, by private endowment and by the larger manufacturing corporations. The war only served to emphasize the importance of such work, so that today scientific research is more generally appreciated and supported than ever before in the history of this country. In Europe the same appreciation exists, but the indispensable material support is, for the moment, comparatively feeble. Whole populations are too concentrated on the problem of finding the next meal to have much energy for work with a deferred, though large, reward.

It is now America's turn to plant the seeds of future progress. For her own sake as well as that of the rest of the world, she must shoulder more and more of the scientific burden. She must expand even her present large research facilities. Her citizens must be brought to an even keener realization of the fact that research pays astounding dividends.

A good example of the right sort of research work is the steam-table investigation now being carried on simultaneously at the Massachusetts Institute of Technology, Harvard and the Bureau of Standards. In giving this work their moral and financial support, engineers and manufacturers are showing their appreciation of the simple fact that it is well worth while to spend a few thousands on the foundation of a structure that is to cost millions. For the steam tables are the very foundation upon which the design, and to some extent the operation, of all steam-generating and consuming apparatus must rest.

The Reliability of Hydraulic Turbines

RELIABILITY of a piece of apparatus is the quality that enables it to render service during regular periods of operation without interruptions and is a property required in all equipment before it can give satisfactory service. Of course no machine can be expected to operate indefinitely without periodic inspection, adjustments and repairs. These will vary with the class of equipment and the duty it is to perform. A steam turbo-generator, which has a large number of small parts, operates at relatively high rotative speeds and is subjected to wide ranges of pressures and temperatures, would in general be expected to require more attention than a hydro-electric unit operating at comparatively slow speed and with few parts to get out of order.

Interruptions to service are not all chargeable to the equipment. A well-trained operating crew will obtain more reliable service from equipment and a plant deficient in design than is obtained from apparatus of superior quality where the attendants are derelict in their duties. This applies not only to inspection and repair of the equipment, but also to the actual operation. A mistake in paralleling exciters may cause not only a complete loss of load from the station, but also serious damage to the direct-current machinery itself. Alternators or systems tied together out of step may result in interruptions to service, damage that will require expensive repairs, and the loss of the apparatus from service for long periods.

Given the proper attention and handled by competent operators, it is generally conceded that hydro-electric machinery is more reliable than steam. This does not mean that a higher degree of reliability can be expected from a hydro-electric plant than from a steam station. The service rendered by a plant or system is largely a matter of operating technique, and a plant or system that may be experiencing considerable trouble with apparatus may be made to give reliable service by proper operation. Large hydro-electric units have a record of over thirty months' continuous operation to their credit, which could hardly be expected from large steam turbo-generators.

While the last few years have seen a number of steam-turbine failures; large hydraulic turbines have also had a number of serious wrecks. In some of the latter the cause was evident, while in others it was not so apparent. However, these failures for the most part have been due to overspeed, obstructions getting into the runner or defective workmanship, causes for which the design of the turbine itself is not responsible. If all the runner failures are considered in comparison with the number of hydraulic turbines in service, they are such a small percentage of the total as to leave no doubt as to the reliability of the hydraulic turbine. However, these failures should not be ignored, for they indicate weakness, in plant design if not in the turbine itself, that if

remedied will improve the reliability of hydro-electric plants. Corrosion of the runner has given considerable trouble in a number of plants, while in others this has been negligible. Corrosion can be overcome by proper design and affects the durability of the turbine rather than its reliability.

Automatic Control for Synchronous Motors

FOR a number of years the tendency has been toward the elimination of the personal element in handling electric machinery. In justification of this there are sound economical and safety-first reasons. First, where the function of a piece of machinery or process depends on the operation of the motor, making the control automatic will not only increase production and reduce costs, but may result in a better product. On the other hand, the human element with all its failings is eliminated. The control can be properly protected from unauthorized persons, and the only feature that the operator has anything to do with is the starting impulse, which may be a push button or similar device. Or, as in many applications, the starting impulse can be given automatically by a float or pressure switch as in the case of pumps or compressors.

For direct-current motors automatic control has been developed to a high degree of perfection. This, with certain exceptions, is true for induction motors, but for synchronous motors automatic-control development might be said to have only been fairly started, since it is comparatively recently that such controllers have become available. This condition has undoubtedly had a retarding effect in the application of this class of motor. Other types of motors could be sold with the assurance to the purchaser that if he so desired the starting could be made independent of the personal element. With the synchronous motor the responsibility for properly starting the machine has been up to an attendant experienced in handling this class of equipment. In these times of labor uncertainty a motor that requires an experienced attendant to perform starting operation must naturally have an element of prejudice against its use when something else can be obtained to do the work, even though the latter may not be quite so well suited to the application, but is automatic in its operation or at least can be started and stopped by the simple process of pressing a button or similar device.

On account of having to excite the motor's fields separately the problem of automatically starting the synchronous motor is somewhat more complicated than that of the direct-current or the induction motor. However, this is one of the best reasons for dependable automatic control for this type of machine—the more complicated the starting operation the greater the necessity for eliminating the human element.

For many years the synchronous motor, as exemplified in the early designs, was a doubtful proposition. Improvement in design has made this class of machine applicable to a great variety of commercial uses that a few years ago were thought impossible. With further improvements in design the motor and control of this class of equipment will become suitable for even a wider field of applications. However, the synchronous machine is like all other types in that its range of applications is going to depend in no small degree upon suitable control equipment, and it is to this end that both the motor and the control manufacturers can well co-operate.

Successful Operating a Matter of Handling Details

STARTING up a prime mover and phasing in on the line is sometimes far from a detail of operation; restoring normal conditions after a plant accident or line trouble may require heroic measures. Operation frequently calls for a quality of engineering far removed from that of handling insignificant details, yet as a matter of fact, success now-a-days depends as much or more on routine work than on the exercise of highly developed skill or judgment in meeting emergency conditions.

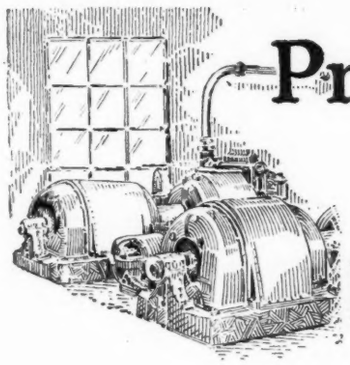
In the old fashioned manufacturing plant an engineer was supposed to correct operating faults when they became annoying. There was ordinarily plenty of time available for repairs which seldom required more than one helper. The engineer's responsibilities may have appeared as heavy as at present, but actually were not.

Several factors have contributed to change power plant conditions. Larger size units require greater time when repairs are necessary. An operating trouble which formerly meant the shutdown of a unit or plant may now mean the loss of load on an extensive transmission system. Continuity of power service and efficiency are both vastly more important than formerly. The result is that there is a greater object in avoiding breakdowns and in keeping the machinery in the best possible condition. The engineer is therefore surrounded with more meters and indicating devices. In addition, control equipment has to a large extent replaced human effort. Details have piled up on every hand.

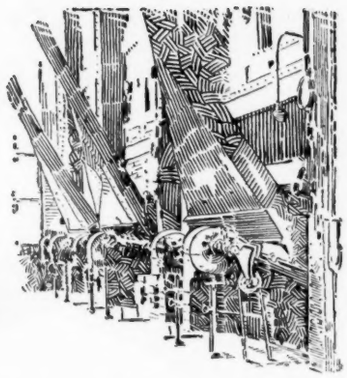
Good operating is required to meet emergency conditions with suitable expedients. It is far better engineering, however, to look ahead and prevent getting into a "jam." The "ounce of prevention" is rising in value. All details must be closely watched in order to insure successful operation. Routine inspections and tests often reveal impending trouble. It may be then a relatively small matter to take preventative measures, compared to the delay that would result by waiting for such trouble to fully develop. Automatic control devices tend to make life more bearable for the operator in many ways, but his maintenance activities and responsibilities have been increased thereby to a large extent over those of a few years ago

Secretary Work's communication to James H. Dunlap, secretary of the American Society of Civil Engineers regarding his dismissal of Arthur Davis from the post of Director of the Reclamation Service gives no real reason and only a poor excuse for such action. Arthur Davis made himself unpopular by insisting that the assessments levied by the Government upon lands benefited by reclamation and irrigation works be paid. Secretary Work says that unless improvements can be made in existing conditions many projects will be abandoned entirely by settlers.

If what is wanted is somebody to bring up successful farmers he had better turn the job over to the Agricultural Department. Some better argument than this will be needed to justify engineering brains in devoting themselves to the government service, or to satisfy the remonstrance of the engineers of the country against this affront to their profession.



Practical Ideas *from* Practical Men



Improving Centrifugal Pump Operation

I recently had occasion to improve a centrifugal pumping outfit. It was a new outfit that had been thoroughly tested at the factory, the pipe lines were of ample size, and the total static head had been accurately measured with surveying instruments before the pump was purchased. In other words, it was not merely a stock pump, but had been built to order and should have given satisfactory results.

When I arrived at the plant, I found the manager about to throw out the centrifugal outfit and install a triplex plunger pump of another make, which meant that I had to either make the pump go or take it home with me. It was half a mile from the pump, which was on the river bank, to the tank at the factory; and the control switch and valves were all at the factory. This meant half a dozen trips back and forth in order to attach the gages and get things in running order.

After the pump was primed and before starting, the pressure gage was read, and this reading was converted into feet. When the height of the gage above river level was added to this, the sum was exactly equal to the static head specified on the order. But when the motor was started, the pressure went immediately to such a point as to indicate excessive friction. The difference was over 20 lb. per sq.in. I recalculated the friction loss in the pipe line and found that it should have been only a little over 3 lb., so I knew there was an obstruction somewhere. The characteristic curve showed that the rate of delivery was only 25 per cent of what was required.

Next I traced the discharge lines, examining all the valves and connections, and then I climbed to the top of the tank, where there was a water-softening plant. The discharge pipe went up under the roof and discharged into the softening tank. I could not see the end of the pipe, but the man in charge told me it opened directly into the tank, and he seemed so positive, I let it go at that.

That evening I tried to figure out what could be causing this shortage of water, and I checked over all the curves and readings, finding no error. I must admit that my conscience bothered me because I had not taken the pains to examine the connection at the softener, so early next morning, I climbed to the top of the tank again. My suspicions were right. There was a float valve on the end of the pipe, which had been in use in connection with the steam pump before the centrifugal pump was installed. In fact, the steam pump was still in service, and as it filled the tank, the float valve would shut off the discharge line and the pump would stop, because the pressure would rise to a point too high for it to act against. I found that this valve was a home-made affair, and never did open to a wide opening. The valve was then removed and the centrifugal pump was started. The water was soon rushing from the end of the open pipe, and I could see at a glance that the pump was up to rating.

I watched the tank fill until the electric float switch shut off the power for the first time. When I got down to the ground, I found the manager smiling. He had been watching a weighted arrow at the end of a rope.

Mt. Clemens, Mich. RUSSELL K. ANNIS.

Backwoods Engineering

At a stave mill on the western shore of Spofford's Lake, in Maine, the power is furnished by a horizontal tubular boiler and a 40-hp. slide-valve engine. The boiler was practically worn out, so to replace it a second-hand, 48-in. by 16-ft. boiler was purchased and I was sent to install it. From the railroad to the mill was six miles of particularly bad road. The boiler was hauled by truck, and when the lower end of the lake was reached the truck broke down and the boiler had

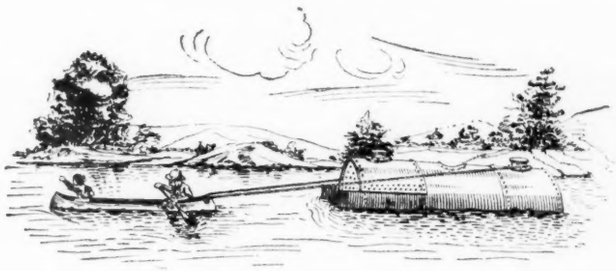


FIG. 1—TOWING THE BOILER TO THE MILL

to be unloaded in the mud at the side of the lake. It was then decided that the best way to get the boiler to the mill, was to "swim" it. Accordingly, the handhole plate was put on (we could not find the yoke to the manhole plate) and the other holes plugged, the boiler was rolled into the lake, care being taken to get it right side up quickly, as the manhole was open. Two woodchoppers who owned a canoe were hired to tow the boiler the half mile to the mill. The wooden plug that had been driven in the blowoff hole was leaking, so that by the time the boiler arrived at its destination, it was half submerged. The bank sloped gently to the water at this point, so the boiler was brought, side-on, to the bank. Stout posts were driven in the ground, tackle blocks were attached, and long ropes were given one complete turn around each end of the

boiler to serve as parbuckles, as shown in Fig. 2; then with a pair of horses attached to the ropes the boiler was parbuckled up the inclined bank. When it was rolled to the end of the rope, chocks were placed to hold it until the rigging was fleeted back, and the rolling was then continued. In four operations the boiler was on the blockings ready for sliding in on the

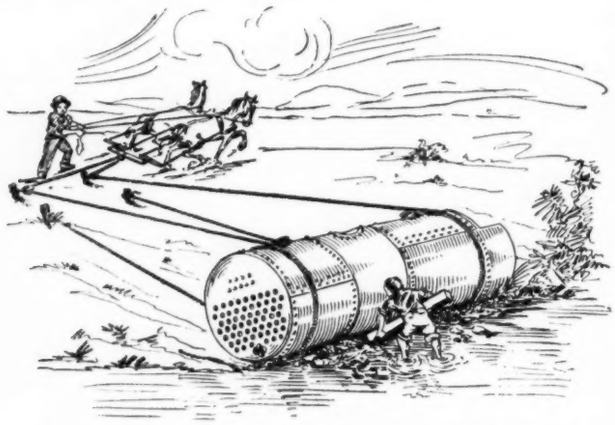


FIG. 2—HOW THE BOILER WAS PARBUCKLED UP THE BANK

foundation. This work required the labor of four men and a team seven hours. The brickwork was completed and the piping connected, and in exactly one week the mill was running.

M. M. BROWN.

Camden, Me.

Obtaining Positive Control of the Damper from the Boiler Front

While trying to improve the combustion conditions in the boiler room, we found that the method of operating the dampers was inadequate because it was difficult to make small adjustments. Various combinations of rods, cables, pulleys, etc., were tried, but they proved unsatisfactory, so we finally devised an apparatus that gave direct and positive control of the draft from the boiler front and at the same time was neat and rugged. The general arrangement is shown in Fig. 1.

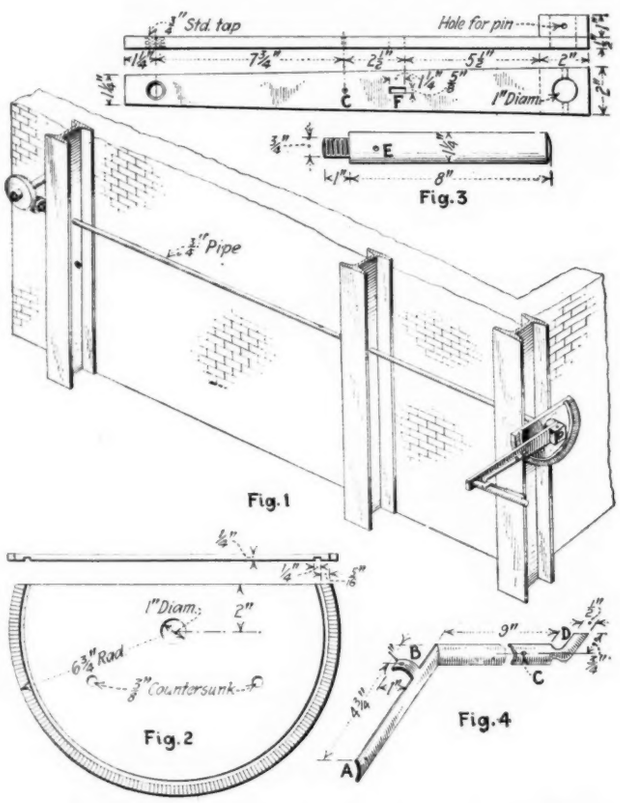
Two miter gears were secured with a ratio of about 3.5 to 1. The lever of the damper was taken off and the large gear put on the damper shaft. The small gear meshed with this and was turned by a handle at the boiler front with a ¾-in. pipe serving as a shaft to connect them. The I-beams which served as buck-stays along the side of the boiler were drilled to support this shaft and serve as bearings. The handle in front moved over a notched sector which held it in position once the draft was adjusted. Everything except the gears was found around the plant, and all the work was done by the regular crew, yet the finished job was neat in appearance and did not require much time to construct.

The small gear was fastened to the end of the pipe by a bolt passing through both the gear and the pipe. At the front of the boiler the pipe passed through the stationary sector and was fastened to the operating arm. This sector is shown in detail in Fig. 2. It was made of ¼-in. flat steel of the dimensions given. After the 1-in. hole was drilled, it was swung in the lathe on a mandrel and the outside edge trued off. Then a groove ¼ in. wide was cut $\frac{5}{16}$ in. from the edge and half the thickness of the plate. Around this outside rim notches were cut about $\frac{1}{16}$ in. wide and triangular in cross-section. This was done with a diamond-point chisel and the notches were afterward smoothed up with a file. The plate was then securely fastened to the I-beam.

The details of the operating arm are shown in Fig. 3. It was made from ½-in. strap iron, cut and drilled as shown. The 1-in. reinforcing pad was riveted on, and then the hole for the shaft was drilled as well as a hole at right angles for a pin to fasten it to the shaft. It was put on the rod with the pad toward the front, and a washer was placed between the arm and the sector to prevent the arm from dragging. The handle is also shown in Fig. 3. It is screwed into the operating arm and clamped with a lock-nut.

This provided the mechanism for turning and adjusting the damper, but to hold it in place a pawl was required. To make this a piece of 1-in. strap was bent as shown in Fig. 4. The projection *B* partly encircles the handle and is pivoted through the hole *E*. The hole is so drilled that when the end *A* is pressed against the handle, the end *D* will be drawn away from the operating arm. The tongue at *D* passes through the opening *F* in the operating arm and engages the notches on the sector.

To keep this engaged, a small coil spring was fastened between the operating arm and the pawl at *C*. This exerted the desired amount of tension.



ASSEMBLY AND DETAILS OF THE DAMPER CONTROL

To operate the damper, it is now only necessary to press the pawl to the handle and turn the arm till the desired amount of draft is shown on the gage, which for convenience is mounted on the post just above the stationary sector.

HENRY L. ROBINSON.

Dallas, Tex.

Comments from Readers

Quick and Ready Methods of Aligning an Engine

Referring to the article by Thos. Airey in the June 26 issue on "Quick and Ready Methods of Aligning an Engine," in the fifth paragraph he says: "In using the connecting rod as a means of testing the alignment, the engineer must be sure that the bore of the large and small end brasses are at right angles with the center line of the rod."

It would be interesting if Mr. Airey would show a simple way in which the engineer could assure himself that the bore of these two bearings is at right angles with the center line of the connecting rod.

Washington, D. C. D. A. MANSON.

What Causes the Pump to Groan?

Referring to the inquiry of P. Stewart in the May 15 issue, as to what causes a pump to groan, a simplex pump, even when new, and more especially if the machining is to a fine finish, will groan owing to the rubbing of the piston rings on the cylinder wall. As wear takes place and the surfaces become polished, groaning will continue and may increase.

A temporary cure, which has been resorted to but is not recommended, is to score the surface of piston rings with a coarse file.

In cases where the groaning is objectionable, such as a feed pump on a private yacht when lying at anchor, the piston rings may be removed and another type substituted. A high-grade asbestos metallic steam gasket can be used, but will require frequent renewals.

New York City. W. J. DENHOLM

Getting Acquainted with the Synchronism Indicator

The article by Ralph Brown in the June 26 issue, on "Getting Acquainted with the Synchronism Indicator," is very complete and interesting, but it does not bring out definitely that the instrument is of the single-phase type and when first connected must be tested out to show the correctness of the position of the pointer.

Assuming that the general wiring is correct so far as can be checked from the diagram of connections sent with the instrument, in my opinion the simplest method for checking the operation is as follows: (1) Disconnect the terminals of one machine at the machine; (2) close the machine breaker; (3) insert the synchronizing plugs, one for the bus or "running" machine and one for the "incoming" machine, which in this case is the circuit to the disconnected machine; (4) the conditions now existing are the same as at synchronism; (5) if the indicator does not assume the correct position—that is, at the top—reverse the low-voltage connections of the potential transformer of the dead machine and verify the result by testing again.

The advantage of this method of testing is in its exact duplication of the true operating conditions and the elimination of the need of any tests for potential transformer polarity.

C. OTTO VON DANNENBERG,
The J. G. White Engineering Corp.
New York City.

Air Tank and Compressor Troubles

Referring to Mr. Lamie's article on "A Double Air Tank Explosion," in the June 26 issue, he does not give any details as to the type and size of the compressors or methods of operating. There is little room for argument about the condition of the metal of the tank, since they were old boiler shells with the tubes removed and new heads put in, one of which was welded, as shown in Fig. 1 of his article. Undoubtedly, the plates in these old shells were more or less crystallized with age, and the process of welding in the heads would not tend to improve the quality of the plate at the joint. Furthermore, no one can tell just how perfect a welded joint it is. For this reason some insurance companies do not approve of long seams or joints being welded unless also riveted. The tanks in question would no doubt carry 50-lb. pressure safely enough, except for gases which are likely to get into the air tanks and under the proper conditions become ignited.

The fact that the bonnet was blown off a valve in the pipe line, as well as a nipple and plug broken, indicates not only the presence of gas in the tank but that it was being carried through the line with the flow of air. These gases are usually caused by the use of a poor grade of oil in the air cylinders. This gas is, of course, formed in the compressor cylinder and discharged into the tank, where it accumulates and is compressed. It is possible, though not very probable, that gas had formed in the compressor cylinders while standing over night, and that upon cutting it into the line the temperature of the air from the other compressor was sufficient to ignite it. Troubles from this source can be avoided by using an oil made especially for lubricating air cylinders and allowing the compressor to turn over slowly a few minutes when starting, with a vent open on the discharge pipe, before opening the valve to the tank. This allows the escape of any gas that may have accumulated, and while running during the day the gas is not formed in such a large volume and also is carried from the tank as the air is used. Air tanks should be blown out at regular intervals to remove any oil which may accumulate in them and which, though liquid, may become gaseous on a day when the compressor is working hard and the air is being carried over at a higher temperature than usual; also, to remove any water that may get into the tanks from leaks in the intercooler or jackets of the cylinders.

We blow out our air tanks about once a week, and though we use a high grade of oil in the cylinders of both our single- and two-stage compound compressors and carry a pressure of 100 lb., we find a thick pasty grease like very heavy cylinder oil, collects in the tank

and also some moisture. In addition to blowing out the tank we clean it every six months, wiping out the thick grease and washing the shell with soda ash and warm water to cut the grease off the plates. We use a vertical tank, hence the accumulation of oil is mostly on the bottom.

It is gratifying to note that some states are passing safety laws for better care of air-pressure tanks.

Cambridge, Mass. R. A. CULTRA.

Reminiscences of a Veteran Mechanical Engineer

Referring to Mr. Odell's account in the July 10 issue, of the Corliss pumping engine for direct pumping at the High Service Station in Providence, R. I., I saw this engine twice and recently found, to my regret, that it had been removed. The room occupied was too valuable to allow it to be retained as an historical relic.

The engine had a vertical shaft, one crank at its upper end, and all connecting rods working in a horizontal plane. Surrounding the shaft there were five steam cylinders and five pump cylinders, alternately, all with their center lines nearly, if not exactly, in one horizontal plane. One of the steam connecting rods worked on the crankpin, but this rod widened out near the pin sufficiently to have inserted in it nine crankpins for the nine other connecting rods. Each steam cylinder exhausted into the condenser.

Mr. Corliss' idea was to have a pumping engine that would move if there was any demand for water, stand still when there was not, and work at any speed between zero and the maximum. Of course its economy was of the lowest order, but when it was built economy in many situations was unimportant.

A few years after the death of George H. Corliss the president of the Corliss Steam Engine Co. was D. M. Thompson, who for many years was manager for B. B. & R. Knight, and who before that was a mill engineer and architect, and associated for a time with Mr. Lockwood, one of the original members of the firm of Lockwood, Greene & Co., now of Boston, Mass.

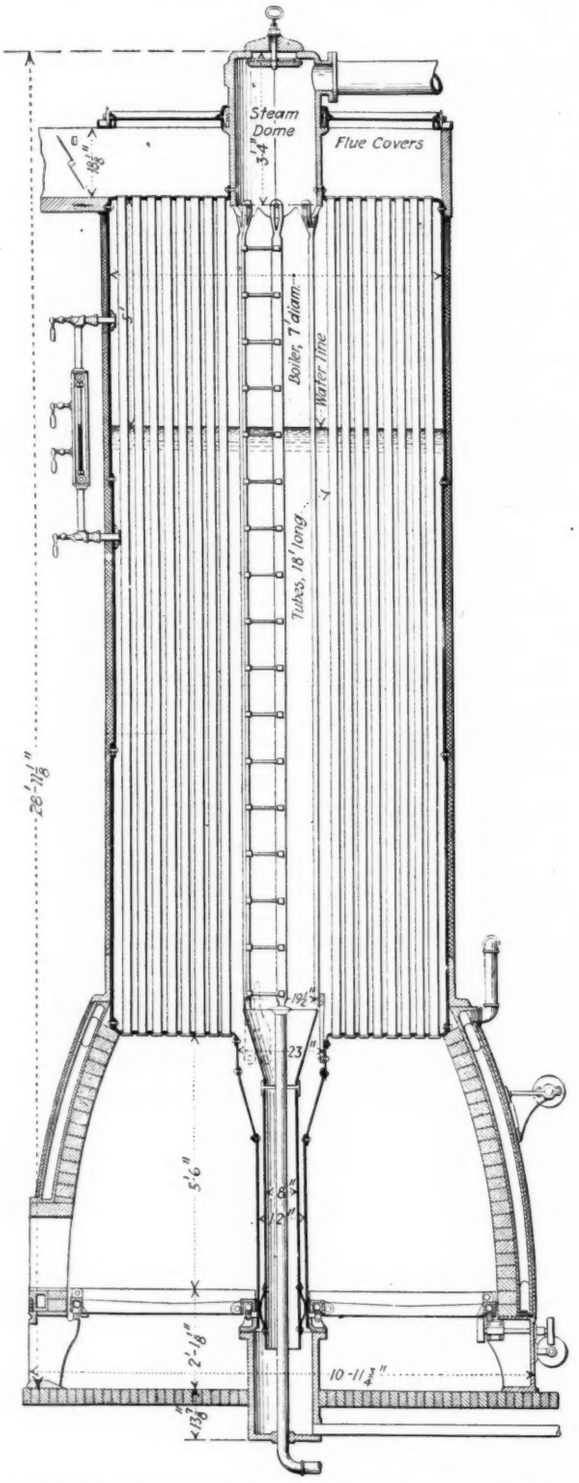
The Corliss Steam Engine Co. manufactured vertical fire-tube boilers, and Mr. Thompson became intensely interested in them. He employed me to make many tests, and it was in this way that I ascertained that vertical boilers are as economical as others.

Mr. Thompson, soon after he assumed the presidency, devised a vertical fire-tube boiler, as shown in the illustration, having the furnace within a cast-iron water-leg shell, which was lined with brick and covered with nonconducting material. The water space was 2½ in. wide, and through this the feed water was pumped at no pressure. This water was delivered into a tank from which it was pumped into the boiler proper by another pump.

Another feature of the boiler was that the grate slowly revolved. There were two firedoors and all parts of the fire were presented to the firemen as they passed, so that any part that was in poor condition could be attended to and the best results obtained. It was expected that most if not all records for economy would be broken, which proved to be the case.

The results of four tests of the boiler are given in Table I, and it will be seen that they were among the best that were ever obtained. The boiler was not forced, as forcing in those days was not done to any extent and was not believed in.

That the economy was really obtained cannot be doubted, as the water was weighed and the error in stopping with the water not quite at the right height, if such was the case, was slight because the water



THOMPSON VERTICAL BOILER FITTED WITH WATER-JACKETED FIREBOX AND REVOLVING GRATES

surface of a vertical boiler is small. The coal was weighed, and as the boiler was heated up by a preliminary fire, which was then drawn, and again started with weighed wood, and the fire burned down at the end, there was no error in the quantity of fuel consumed.

We then have correct fuel and correct water. The gases were analyzed frequently.

The boiler possessed qualities that necessarily produced good results. The radiation was next to nothing around the firebox, the escaping gases were utilized in superheating the steam, and the firing was of the very best, as the gas analyses showed. The boiler had been tested by Mr. Odell several times before I began my tests, and the fireman had been carefully trained.

I regret that I did not take quite enough data to construct a heat balance, for some people may think that a boiler cannot give such high efficiencies as here reported.

TABLE I—TESTS OF THOMPSON VERTICAL BOILER WITH WATER-JACKET FIREBOX AND REVOLVING GRATE

Kind of coal, New River run of mine.	Kind of tests, standard fresh wood fire.			
Date, 1896, Sept	10	11	23	24
Duration, hours	11	11	11	9
Grate area, sq.ft	67.94	67.94	67.94	67.94
Water-heating surface, 2,458 sq.ft. superheating surface, 966 sq.ft				
Total heating surface, sq.ft	3,424	3,424	3,424	3,424
Gage pressure, lb	92.00	91.10	83.60	84.44
Atmospheric pressure, lb	14.72	14.89	14.92	14.96
Absolute pressure, lb	106.72	106.00	98.52	99.40
External air temp., deg. F	84	90	53	63
Fireroom temp., deg. F	72	90	76	83
Feed-water temp., deg. F	69.5	70	67	66
Escaping gases temp., deg. F	397	416	386	383
Moist coal, lb. per hour	873	709	725	800
Moisture in coal, per cent	6.2	3.1	2	2
Dry coal, with wood equiv. added, lb. per hour	830	698	721	802
Dry refuse per hour, lb	57.5	53.2	43.6	40.7
Dry refuse, per cent	6.9	7.6	6.05	5.07
Combustible per hour, lb	773	645	677	761
Superheat, deg. F	33	41	36	37
Heat value of one lb. of dry coal, B.t.u	13,884	14,041	13,782	13,782
Heat value of one lb. of dry combustible, B.t.u	14,781	14,786	14,455	14,455
Water supplied to boiler per hour, lb	8,411	7,208	7,292	7,780
Steam generated corrected for superheat, lb	8,539	7,338	7,418	7,917
Factor of evaporation (incl. superheat)	1.204	1.208	1.206	1.206
Water evap. per lb. of dry coal, (incl. superheat), lb	10.29	10.51	10.29	9.87
Equivalent evap. from and at 212 deg. lb	12.39	12.70	12.41	11.93
Water evap. per lb. of combustible (incl. superheat), lb	11.06	11.38	10.95	10.40
Equivalent evap. from and at 212 deg., lb	13.31	13.75	13.21	12.57
Efficiency of boiler based upon dry coal, per cent	86.59	87.77	87.60	84.01
Efficiency of boiler based upon dry combustible, per cent	87.39	90.22	88.69	84.38
Dry coal consumed per sq.ft. grate per hour, lb	12.31	10.35	10.70	11.89
Dry coal consumed per sq.ft. heating surface, per hour, lb	0.338	0.284	0.294	0.234
Water evap. from and at 212 deg. per sq.ft. heating surface, per hour, lb	3.002	2.589	2.613	2.793
Water evap. from and at 212 deg. per sq.ft. grate, per hour, lb	151.31	130.47	131.68	140.77
On basis 34½ lb. water per hr. from and at 212 deg., hp	298	257	259	257
No. of sq.ft. of heating surface per hp	11.49	13.32	13.22	12.36
Horsepower per sq.ft. of grate area	4.39	3.78	3.81	4.08
Horsepower at 12 sq.ft. of heating surface, incl. superheating surface	285	285	2.85	285

TABLE II—RESULTS OF TESTS OF THE INGLIS SCOTCH BOILER BUILT BY INGLIS BOILER SYNDICATE, GLASGOW, SCOTLAND

Tests by	Coal Burnt per Sq.ft. Grate per Hour Lb.	Heat Value of Coal, B.t.u.	Temperature of Escaping Gases, Deg. F.	Evap. per Lb. of Coal From and at 212 Deg. F., Lb.	Efficiency Per Cent
Bustall & Monkhouse	19.82	14,350	439	12.97	87.34
	27.31	14,280	482	12.24	82.80
	48.89	13,890	523	11.33	78.80
Professor Stanfield	20.10	14,080	457	12.70	87.58
	33.20	14,080	475	11.82	81.32
	48.00	13,600	524	10.94	77.82
John Brown & Co., Ltd.	38.70	13,647	498	12.08	85.00
	48.90	13,647	593	11.40	80.20
Denny & Co.	16.20	13,864	470	12.87	89.50
	19.00	13,864	410	12.70	88.40
	29.00	13,864	440	12.08	84.10
The Fairfield Shipbuilding & Engineering Co. Ltd.	19.20	13,952	445	12.45	86.21

There is one possible source of slight error. It was my custom in those days to have the coal subjected to ultimate analysis and to compute the heat value by Du Long's formula. Nowadays, tests in a coal calorimeter are always used, but fortunately I also, in many cases had coal tested in a calorimeter as well as analyzed, and therefore have means of determining the ratio between the two. It is almost always 0.93½, and this ratio I used in working up the tests of the Thompson boiler recently. I also used the Marks and Davis tables in the recent computations, which produced results higher than those originally obtained, when I used Peabody's tables, fifth edition.

Concerning the possibility of as high efficiencies as here given, I will quote from reports by several English engineers who tested an Inglis Scotch boiler. Some of these names are so eminent that it is difficult to disbelieve their results, but they gave nothing in their reports that enable a person to check them. These results are given in Table II.

F. W. DEAN,
Boston, Mass. Wheelock, Dean & Bogue, Inc.

Dual-Clearance Unaflow Engine

I was interested in the article on "A Unaflow Engine Without Auxiliary Exhaust Valves," which appeared in *Power*, Dec. 26, 1922. I gathered from the description given that the dual clearance pipe used in the Harrisburg engine is not only a substitute for the extra clearance spaces in the cylinder head and for the auxiliary exhaust valves, but is a novel and efficient means of avoiding the objectionable features of both these methods for preventing excessive compression in unaflow engine cylinders when working non-condensing.

Now comes Benjamin T. Allen's letter in the issue of July 17, in which he states that while "for condensing operation the cylinder clearance between the valve and piston is made of the least volume possible, for non-condensing operation larger clearance is required than when the engine is run condensing, so to provide for this condition pockets are cast usually on the sides of the cylinder near the end flanges. . . . Communication is established with the pocket and cylinder clearance by a port that can be opened or closed as may be required. This is usually accomplished by a valve held in its seat by a spring which opens under the increased back pressure in the cylinder should the vacuum break." He also adds that "the action of the dual clearance goes on just the same, whether the engine is running non-condensing or condensing."

Granted, but of what service is the dual clearance, when the older arrangement of additional clearance spaces in the cylinder has to be resorted to?

If the dual clearance pipe can do what appears to be claimed for it, why is it necessary to bring out a pure Stumpf unaflow engine to be used entirely for condensing operations except in cases of emergency?

Referring to the indicator diagrams from a 17x16 Harrisburg dual-clearance unaflow engine running non-condensing, a glance at the compression line would lead me to infer that the clearance volume is nearer 20 than 5 per cent, as is the case in some unaflow engines using auxiliary exhaust valves.

Large clearance volume is incompatible with low steam consumption.

DUNCAN A. MORTON.
El Segundo, Calif.

The steam consumption of Curtis turbines is now about one-half of that of an early turbine, placed in operation in 1903. The efficiency of the turbine generator unit has been increased from 47 or 48 per cent in 1903 to 75 or 78 per cent in 1920. Recently improved steam conditions have further decreased the water rate.

QUESTIONS AND ANSWERS

Conducted by Franklin VanWinkle

Breakage of Firebox Staybolts

What is the cause of breakage of firebox staybolts? J. R. M.

The chief cause for breakage of the ordinary forms of staybolts is unequal expansion of the connected sheets. One of the sheets usually is in contact with the fire, and frequently, from lack of circulation of the boiler water, the other sheet does not attain the general temperature of the boiler. This difference in temperature of the sheets gives rise to considerable relative motion between the sheets with the result that rigid staybolts are likely to become cracked or there is leakage at the fastenings.

Temperature of Water Cooled By Ice

If 20 lb. of ice at 32 deg. F. is melted in 60 lb. of water of the initial temperature of 60 deg. F., what will be the final temperature of the water? T. C.

The heat lost by the water would be equal to the heat gained by the ice. In melting to water at 32 deg. F. each pound of the ice would receive the latent heat of fusion, or 143 B.t.u., and in cooling 60 lb. of water from 60 deg. F. to 32 deg. F., the loss of heat by the water would be $(60 - 32) \times 60 = 1{,}680$ B.t.u. Hence the water would be cooled to 32 deg. F. by the melting of only $1{,}680 \div 143 = 11.748$ lb. of ice, so that the mixture would consist of $60 + 11.748 = 71.748$ lb. of water at 32 deg. F. and $20 - 11.748 = 8.252$ lb. of unmelted ice.

Grounding the Neutral of Alternating-Current Generators

Where alternating-current generators are equipped with differential relay protection, is it general practice to ground the neutral of the generator? If so, is the ground made directly or through a resistance, and is a switch placed in the circuit? M. E. T.

It has become general practice to provide differential protection for large alternators so as to disconnect them from the circuit in case of trouble developing in the machine's windings, and to ground the neutral of the machine just as if the protective relays were not used. Unless the neutral was grounded, the machine would not be tripped off the circuit in case of an insulation failure to ground on one phase. It would require a failure between phases before the generator would be disconnected with an ungrounded neutral. A resistance is connected between the ground and neutral to limit the current to a reasonable value until the relays have disconnected the generator.

To connect the neutral to the ground, a non-automatic oil switch is used. Where two or more generators are installed in a station, which is generally the case, a ground bus is used to which the generators may be connected through a non-automatic oil switch, it being general practice to have but one generator connected to the ground bus at one time. In some cases provisions have been made for the differential relays to open the ground oil switch in case a generator is tripped off the bus owing to an internal fault. A feature that is coming into general use is to connect in the ground circuit a current transformer which is used to operate a relay and light a signal lamp in case the current to ground exceeds a predetermined value. As the primary purpose of differential protection is to disconnect the generator from the line in case of internal trouble in its windings, it is also essential to have the field circuit opened when the generator oil switch opens, otherwise the machine will continue to supply power into the fault.

Extraordinary Stresses Induced by Rupture

When a boiler explosion occurs, how is it explained that joints and other parts, computed to be stronger than necessary to withstand the boiler pressure carried at the time, are found to be ruptured? M. O. N.

When one part gives away, other parts may become burdened with greater stresses than when the structure, as a whole, is in equilibrium. Extraordinary stresses thus induced usually result in a train of failures that occur in rapid succession, and generally there are evidences that point to the fact that many parts have become ruptured in this manner after the exploded boiler has become moved from its original position.

Cleaning Grease from Gage Glass

What is a good way to clean a water-gage glass on a low-pressure boiler without removing the glass tube? P. M. C.

When the boiler is steaming under 5 to 10 lb. pressure, draw a cupful of hot water from the boiler and add about a tablespoonful of muriatic or other acid. Close both water-gage valves, then open the pet cock at the bottom and blow all water out of the glass by opening the top valve, and immediately close the top valve and submerge the end of the pet cock in the cup of hot-water solution. A vacuum is at once formed in the gage glass, causing the solution in the cup to rise and fill the glass. Keep the pet cock immersed and by slightly opening and closing the top valve, allow the solution to be alternately expelled and drawn into the glass until all grease, oil or other matter adhering to the inside of the glass is cut out. Then close the pet cock and resume regular operation of the water gage by opening both water-gage valves.

Temperature in Coils Supplied by Steam Through Reducing Valve

When dry saturated steam at 125 lb. gage is passed through a reducing valve and discharged to steam heating coils, what is the temperature of the steam in the coils? A. R. B.

According to Marks and Davis steam tables, a pound (weight) of steam at the pressure of 125 lb. gage, or 140 lb. per sq.in. absolute, contains 1,192.2 B.t.u. above 32 deg. F., and a pound of dry saturated steam at 2 lb. gage, or 17 lb. per sq.in. absolute, contains 1,153.1 B.t.u. above 32 deg. F. As there is neither gain nor loss of total heat from passing the steam through the reducing valve, there will be 1,192.2 — 1,153.1 = 39.1 B.t.u. more than necessary for a dry saturated condition of the steam at the reduced pressure.

By reference to tables of properties of superheated steam, it may be found that a pound of steam at 2 lb. gage, or 17 lb. per sq.in. absolute, containing 1,190.8 B.t.u. would have a temperature of 299.4 deg. F., and containing 1,195.5 B.t.u. would have a temperature of 309.4 deg. F. Therefore, by interpolation the temperature of the steam immediately after passing through the reducing valve would be $\frac{309.4 - 299.4}{1{,}195.5 - 1{,}190.8} \times (1{,}192.2 - 1{,}190.8) + 299.4 = 302.4$ deg. F. However, the 39.1 B.t.u. of superheat is only a small proportion of the total heat, and as the superheating is the first to be liberated, under average conditions the temperature of the steam pipes and coils a short distance beyond the reducing valve rarely is higher than the temperature of dry saturated steam corresponding to the pressure, and for 2 lb. gage this would be 219.4 deg. F.

Packing Joint at End of Through Stay

We are troubled with leakage through the flue sheet at the end of a through stay rod of our horizontal return-tubular boiler. Several attempts have been made to stop the leakage by removing the nut and repacking around the stay with asbestos wicking and white lead and also with a copper ring, but this only answered for a short time. What remedy is suggested? F. A. W.

The amount of leakage depends on the kind and permanency of joint made between the screw threads of the staybolt and sides of the hole in the flue sheet. Recurrence of the leakage was probably due to the presence of rust or pieces of old packing left in the space around the screw thread.

Remove the outside nut and washer and after backing off the inside check-nut, clean the screw thread and hole of all loose dirt and wash out the space with turpentine. Then pack the space around the bolt from inside as well as outside of the boiler with short ravelings of asbestos and white lead ground in linseed oil. The packing can be tucked into place with a flattened piece of wire. Temporarily set up the outside nut and washer and after doing a good job of packing from the inside of the boiler set up the inside nut with a firm metal bearing against the flue sheet and afterward slacken off the outside nut and see that the packing is filled from the outside so it will be compressed and fill the whole packed space when the outside nut and washer are firmly set in place with metal-to-metal bearings so there will be no opportunity for disturbance of the packing from movement of the staybolt in the hole of the tube sheet.

Strength of Diagonal Joint

Knowing the efficiency of the type of joint used as a diagonal joint on a cylindrical boiler shell, what is the method of determining the efficiency required for a longitudinal joint capable of resisting the same boiler pressure? R. L. F.

In case the angle *ACB*, Fig. 1, which the line of the joint makes with the center line of the boiler is less than 45 deg., the strength of a diagonal joint *AC*, Fig. 1, taken along a longitudinal line *BC* can be determined approximately by multiplying the efficiency of the joint by its length *AC* and dividing by the length *BC*. When the angle *ABC* is 45 deg., the error is about 3 per cent, and for smaller angles the error is less, for 30 deg. amounting to only about ⅛ of 1 per cent. Thus if the efficiency of the seam *AC* computed in the usual manner were 55 per cent and the angle of *ACB* such that *A* to $C = 28$ and *B* to $C = 20$ in., then $0.55 \times \frac{28}{20} = 0.77$, or 77 per cent, would represent approximately the efficiency required of a longitudinal joint for resisting the same pressure as a joint of 55 per cent efficiency at the angle *ACB*. While this method of calculation is close enough for most practical purposes, there is an inaccuracy introduced, as no account is taken of the stress lengthwise of the shell.

If greater accuracy is desired, the strength of joint placed diagonally compared with one parallel to the

FIG. 1—ORDINARY DIAGONAL PATCH-JOINT

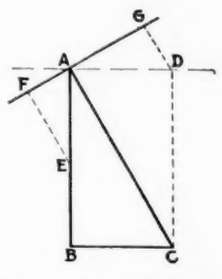


FIG. 2—GRAPHICAL METHOD OF RESOLVING DIAGONAL JOINT

center line of the shell can be determined graphically as follows:

Referring to Fig. 2, lay off any convenient distance *AC* along the joint; draw *AD* and *BC* parallel with the center line of the boiler and *AB* perpendicular to *AD*. Through *A* draw *FAG* at right angles to *AC*, to represent the direction of resistance of the joint; make *AD* equal to *BC*; lay off *AE* equal to one-half of *AB* and draw *DG* and *EF* perpendicular to *FG*.

Then for any stated pressure within the boiler, the stress per unit of length created along a longitudinal line or joint *BC* would cause $\frac{FG}{AC}$ times as much stress per unit of length of a joint *AC* and the joint *AC* could resist $\frac{AC}{FG}$ times as much pressure as if used as a longitudinal joint *BC*.

Assuming that $AC = 25$ and $FG = 16$ and the efficiency of the joint *AC* is 0.55 of solid plate, then the strength would be sufficient to resist the stress that could be brought on a longitudinal joint whose efficiency was $\frac{25}{16} \times 0.55 = 0.853$, or 85.3 per cent of the strength of the solid plate.

Easy Lessons in Engineering

Figuring Engine Efficiencies

Take the case of a steam engine and generator unit, for which a test gives the following data:

Power output	83 kw. / 124 b.hp. / 135 i.hp.
Hourly steam consumption	3,240 lb.
Steam pressure at throttle	150 lb. gage
Steam superheat at throttle	100 deg.
Exhaust pressure	2 lb. gage

These data are sufficient for working out most of the efficiencies commonly used. Considering the engine as a complete unit by itself, its output is 124 b.hp. (brake horsepower). The 135 i.hp. (indicated horsepower) is the power delivered to the piston by the steam.

The output of the generator, and also of the engine and generator considered as a single unit, is 83 kw. Since 0.746 kw. equals one horsepower, this may be put in the form of $83 \div 0.746 = 111$ e.hp. (electrical horsepower) to simplify the comparison with the brake and indicated horsepower. Subtracting 111 e.hp. from 124 b.hp. we find that 13 hp. are lost in the generator. Next comes what is called the "mechanical efficiency" of the engine. To understand this, picture the mechanism of the engine by itself. Then the input of the mechanism is seen to be the 135 i.hp., which the steam supplies to the piston, while the output is the 124 b.hp. which the coupling delivers to the generator. We then have:

$$\text{Mechanical efficiency of engine} = \frac{\text{output}}{\text{input}} = \frac{124}{135} = 0.92 \text{ or } 92 \text{ per cent.}$$

More important than any of these is the "thermal efficiency" which may be based on the indicated, brake or electrical horsepower. These three thermal efficiencies are really the over-all efficiencies up to the piston, the coupling and the switchboard, respectively. To simplify the computations we may first figure the steam consumption per horsepower for each of the three cases as follows:

$$\text{Steam consumption per e.hp. hour} = \frac{3,240}{111} = 29.2 \text{ lb.}$$

$$\text{Steam consumption per b.hp. hour} = \frac{3,240}{124} = 26.1 \text{ lb.}$$

$$\text{Steam consumption per i.hp. hour} = \frac{3,240}{135} = 24.0 \text{ lb.}$$

So we can take the output as 2,545 B.t.u. (the heat equivalent of one horsepower hour) in each case, provided we use the input corresponding to the particular rate of steam consumption just figured. The next step is to figure the heat input corresponding to each case. At this point it is easy to fall into a common error. The total heat in a pound of steam at 165 lb. absolute pressure (150 lb. gage) and 100 deg. superheat is given in the steam tables as 1,252 B.t.u. It seems plausible that this quantity multiplied by the weight of steam supplied should be the input. Actually this is wrong. If, for producing a certain result, you give a man $100 and he later returns $10, the input of money for the job is not $100 but $100 — $10 = $90. In the same way the heat input of the engine is the heat in the steam supplied minus the heat returned.

But this raises still another point. Shall the engine be credited with all the heat in the exhaust or only the part that can be returned to the boiler? By general agreement the latter is used whenever thermal efficiency is figured.

Imagine then that the entire exhaust of the engine passes to a perfect feed-water heater that handles one pound of feed water for every pound of exhaust steam. The temperature of exhaust steam at 17 lb. absolute (2 lb. gage) is found from the steam tables to be 219 deg. and the heat of water at the same temperature to be 187 B.t.u. It is then evident that for each pound of steam received the engine could return to the boiler, with the aid of a perfect feed-water heater, 187 B.t.u. in the form of water at 219 deg. The actual heat in the exhaust steam is far more than sufficient to heat the feed water to 219 deg., but it is impossible for the steam to heat the water hotter than itself, so the excess must blow to waste and is, from the point of view of the feed-water heater, never returned. So the standard practice is to charge the engine with the difference between the total heat of the throttle steam and the heat in *water* at a temperature corresponding to the exhaust pressure.

In this case, then, the input per pound of steam is 1,252 — 187 = 1,065 B.t.u. The three thermal efficiencies may now be computed as follows:

Thermal efficiency based on electrical output:

$$\frac{2,545}{29.2 \times 1,065} = \frac{2,545}{31,100} = 0.082, \text{ or } 8.2 \text{ per cent.}$$

Thermal efficiency based on brake output:

$$\frac{2,545}{26.1 \times 1,065} = \frac{2,545}{27,800} = 0.0915, \text{ or } 9.15 \text{ per cent.}$$

Thermal efficiency based on indicated output:

$$\frac{2,545}{24.0 \times 1,065} = \frac{2,545}{25,600} = 0.0995, \text{ or } 9.95 \text{ per cent.}$$

In addition to these efficiencies, two others are sometimes computed. One of these is the thermal efficiency of a perfect engine for the same initial steam pressure and condition and the same exhaust pressure. This is called the "efficiency of the Rankine cycle." It is of course always higher than the actual thermal efficiency. To compare the actual thermal efficiency (any one of the three) with that of the perfect engine for the same conditions the former is divided by the latter. The result is called the "efficiency ratio" or the "Rankine efficiency" and is a measure of the degree to which the actual engine approaches the theoretical possibilities. To figure the efficiency of the perfect engine is rather a long computation involving the use of entropy, so this will be reserved for another article.

Fallacies Concerning Power Development*

By O. C. Merrill†

There are going on at the present time in the field of power development and of public-utility service in general, many controversies that are hindering the development of our resources and delaying the extension of utility service. There is also much popular ignorance of what is involved in the way of capital outlay and of technical ability in developing and operating modern electric power systems and of what are the elements that enter into the cost of service. This popular ignorance is due primarily to lack of correct information and is the fundamental cause of the majority of the existing controversies. More unwise popular movements are started and maintained because of ignorance of the facts than from all other causes combined. The public is being constantly supplied with false information and flooded with propaganda by individuals and agencies seeking to profit politically or otherwise at their expense. The most valuable public service which, in my opinion, this conference could render would be the creation of some agency which would have the means for making a study of the power situation within this territory, which would be able to distinguish between facts and fancies, and which would have the courage to present an intelligent and unbiased interpretation of the information secured.

As indicative of the need of some such intelligent and disinterested source of information I propose to discuss today certain popular fallacies concerning power development, its cost, management and control.

The first of these is: Since water costs nothing and runs down hill of itself, water-power development is a relatively simple and inexpensive matter, which almost anyone is capable of undertaking. At least many of the applications filed with the Federal Power Commission would indicate the prevalence of such a point of view; for we have had applications from individuals, from corporations and from municipalities, which present no showing of ability to finance the proposed undertaking, which give evidence of no training or experience in hydro-electric development, and which plainly indicate that the applicant has no adequate conception of the problems involved in modern water-power development.

Lack of training and ability appear not only in the design but also in the estimates. In computing the capital cost of a power development too many people stop at the generator switchboard and overlook or ignore the costs of transmission and distribution and of auxiliary steam reserve. Too many, also—and I must include in this list many so-called engineers—fail to take account in their estimates of accessory construction costs.

As illustrating the major cost elements in a typical modern hydro-electric plant certain figures of actual cost from the Commission's files of a plant recently completed in California may be cited. This plant of 45,000-hp. capacity consists of a diverting dam, a tunnel some 3½ miles in length, a concrete power house with generating equipment, 180 miles of transmission line and four substations. The total cost of the plant up to the point of distribution is $156 per hp. Of this amount the power-house with all its hydraulic and electrical equipment cost only $29 per hp., less than one-fifth of the total. The dam, conduits, and tail-race together cost $89, and the transmission line and substations, $34. Transportation facilities alone cost nearly $5 per hp. Furthermore, if this plant were not merely an addition to an existing system, it would be necessary to add from $40 to $60 for steam reserve and probably as much more for distribution line and equipment.

The danger of overlooking essential cost elements may be illustrated by the estimates for a certain project on the Colorado River which has been proposed for development for the purpose of supplying power to the adjacent territory and to the municipalities of southern California. In fact, the City of Los Angeles had on its ballot in a recent election a proposal, which was, however, defeated, to bond the city for $35,000,000 in order that it might enter upon this undertaking. This project has been given wide publicity and claims have been made that it will produce unusually cheap power. The original project contemplated the construction of a dam some 700 ft. in height, necessitating foundations more than 100 ft. below the river bed and creating some 600,000 hp. No accurate estimates of the cost of such a development appear to have been made; at least, of costs up to the points of energy distribution, which in this instance would vary from 100 to 300 miles from the power plant. A careful study has, however, been made by the Commission's engineering staff of a dam of one-half the proposed height, and developing 225,000 hp., all that would appear feasible at the present time. The estimated cost of dam, power-house, generating equipment and step-up transformers is $28,800,000, or $128 per hp., while the cost of transmission, including six substations, is $29,800,000, or $132 per hp., making a total of $260, or $104 per hp. more than the previous example cited. Furthermore, the estimates assume that the energy will be delivered to distribution systems already in existence and that, therefore, nothing need be included for distribution lines and equipment or for steam auxiliaries.

Electric-power development, and particularly hydro-electric development, if it is to make the advances necessary to meet the increasing demands of industry, will require enormous amounts of new capital, far beyond the financial capabilities of any small group or of any single community.

A popular misconception similar to the one I have just discussed arises from the failure to distinguish between operating costs computed at the generator switchboard and such costs computed at the consumers' meters. Furthermore, because water costs nothing, it is assumed that hydro-electric power must necessarily be much cheaper than steam power. Water is cheaper than fuel, it is true; but a hydro-electric plant with all its accessories may cost several times as much as a steam plant of equal capacity, and in addition may require an expensive transmission line, substations and steam reserve.

Distribution of Costs in California

Studies made by the California Railroad Commission in a rate case before that body are illuminating as indicating the variation in cost of energy from the hydroelectric plant to the retail consumer. The figures are actual average costs to one of the large California electric utilities, such costs including operating expenses, depreciation and return on investment for each part of the system considered. The average energy cost at the generator switchboard is 4.25 mills, less than one-half cent, per kilowatt-hour. To guarantee continuous service and to furnish part of the energy supply in periods of low water, it is necessary to add operation, depreciation and interest on steam reserve plants, thus increasing the cost to 6.28 mills. Transmission losses and costs, taxes and general expenses bring the average cost of power at the substation up to one cent per kilowatt-hour. The distribution costs from the substation to the agricultural or industrial consumer having an installation of about 20 hp. practically double the cost, making his charge two cents per kilowatt-hour. When we consider the domestic lighting consumer, however, we find that his cost is primarily in the local distribution; that is, in interest, depreciation and maintenance on the distribution system and meters, in the expenses of handling consumers' accounts, and in taxes and other general expense items chargeable to this class of service.

The California studies show that while the cost at generator switchboards is only 4.25 mills, at the substation only one cent, and when delivered to the general power consumer slightly less than two cents, the cost of delivery to the ordinary residence consumer is 7.4c.; and that of this aggregate cost, less than 6 per cent is chargeable to hydro generation, while 88 per cent occurs between the substation and the consumers' fixtures.

*Abstract from a paper prepared for convention of Southern Appalachian Water-Power Conference, Asheville, N. C., June 27, 1923.

†Executive secretary, Federal Power Commission.

News in the Field of Power

Enrollment of Engineering Students Shows Decrease

Walton St. John, specialist in rural and technical education, of the United States Bureau of Education, has collected educational statistics from 129 engineering schools for the years 1921-22 and 1922-23. The enrollments have been obtained for the first semester of each year, and tabular summaries of these enrollments have been published. With these figures it has been possible to ascertain the enrollments of the graduating classes in the several branches of engineering and also to predict with some degree of assurance the enrollment of the graduating classes for the year following.

Loss of students in the five leading branches of engineering in 1922-23 from 1921-22 based on the enrollments during the first semester of the years are recorded as follows: Civil engineering, total enrollment 12,802 with a loss from the previous year of 1,590; mechanical engineering, total 14,453, with a loss from the previous year of 1,561; electrical engineering, total 13,275, with a loss from previous year of 644; chemical engineering, total 7,054, with a loss of 1,668 from previous year; mining and metallurgical engineering, total 2,895, with a loss from the previous year of 234.

Knoxville Power & Light Co., Plans Two More Developments

To add interest to the controversy going on between the Knoxville Power & Light Co., the Tennessee Electric Power Co. and the Tennessee Hydro-Electric Co. as related in the July 31 issue of *Power*, the Knoxville Power & Light Co. has amended its application to the Federal Power Commission, for four power projects on the Clinch and Powell Rivers, Tennessee, so as to add two developments on the Tennessee River proper above the mouth of the Clinch River and below the City of Knoxville. It proposes to build a dam 53 ft. high at the so-called Coulter site in Blount County, developing 48,000 hp., and to develop 44,000 hp. by a 34-ft. dam at the Marble Bluff site, a short distance below. This will give a total contemplated development in the six sites of nearly 300,000 hp.

World's Power Conference Will Be Held in London in 1924

American Participation Is Now Being Arranged For In London By General Chairman O. C. Merrill

IN CONNECTION with the British Empire Exhibition to be held at Wembley, London, next summer, there will be held during July, a World Power Conference, which is being arranged by The British Electrical and Allied Manufacturers' Association, whose director, D. N. Dunlop has offices at 36 Kingway, London, W. C. 2. It is hoped that the conference will be participated in by many technical and scientific institutions and industrial organizations in Great Britain and other countries.

The object of the Power Conference will be to consider how the industrial and scientific sources of power may be adjusted nationally and internationally: By considering the potential resources of each country in hydro-electric power, oil and minerals; by comparing experiences in the development of scientific agriculture, irrigation and transportation by land, water and air; by conferences of civil, electrical, mechanical, marine and mining engineers, technical experts and authorities, and industrial research; by consultation of the consumers of power and the manufacturers of the instruments of production; by conferences on technical education to review the educational methods in different countries, and to consider means by which existing facilities may be improved; by discussions on the financial and economic aspects of industry; by conference on the possibility of establishing a permanent world bureau for the collection of data, the preparation of inventories of the world's resources and the exchange of industrial and scientific information through appointed representatives in the various countries.

The proposed sections of the conference are as follows: I—Power Resources; II—Power Development; III—Power Applications; IV—Power, Economic and Financial.

In response to the invitation for American participation, twenty-odd engineering, technical and industrial organizations have appointed representatives to the committee of arrangements which has planned the following organization: Hon. chairman, John W. Weeks, Secretary of War; hon. vice-chairmen, Hubert Work, Secretary of the Interior, H. C. Wallace, Secretary of Agriculture, Herbert Hoover, Secretary of Commerce, Owen D. Young, Samuel Insull, Prof. Lester Paige Breckenridge, Sydney Z. Mitchell; general chairman, O. C. Merrill; vice-chairman, H. J. Pierce; secretary, W. M. Steuart; treasurer, H. M. Addinsell; executive committee, Peter Junkersfeld, representing the A.S.C.E.; Calvin W. Rice, the A.S.M.E.; David P. Rushmore, the A.E.C.S. and the A.I.M.M.E.; Calvert Townley, the F.A.E.S.; H. I. Harriman, the United States Chamber of Commerce; Fred R. Low, the A.S.M.E. and the F.A.E.S.; John W. Lieb, the A.I.E.E. and Associations of Edison Illuminating Companies.

Working committees have been appointed and General Chairman Merrill is now in London gathering information as to the scope and details of our expected participation.

Invitations will be extended to other organizations who may be interested and it is the purpose and hope to make a contribution to this first effort toward a systematic study of the world's power resources and their utilization commensurate with the part America has played in the development of this "age of power."

Power of St. Louis River To Be Fully Developed in 1924

When the American Light & Power Co., an Eastern corporation, purchased the holdings and riparian rights of all hydro-electric companies on the St. Louis River in northeastern Minnesota, it marked one of the most important steps in industrial development in northern Minnesota since the opening there of the greatest iron mines in the United States.

By the autumn of 1924 the St. Louis River, which rises in the northern part of the state and flows into Lake Superior near Duluth, will perhaps be the only river in the country to have its available power fully harnessed.

The purchase included the holdings of the Great Northern Power Co., the Duluth-Edison Co. and the Minnesota Utilities Co., all of Duluth, Minnesota; the General Light & Power Co., and the St. Louis River Power & Improvement Corp., of Cloquet, Minn. These consist of seven storage basins, most of the retaining dams now completed, and seven hydro-electric generating stations, five of which are now in operation, one in the course of construction, and one to be completed next year. An additional storage dam will be constructed during 1924 which will stabil-

ize the flow of water the year round, developing a continuous 134,000 hp.

From Cloquet to St. Louis Bay on Lake Superior, the river has a fall of 578 ft. Hydro-electric plants which are now in operation and which will be completed during the coming year will have a combined water head of 570 ft., leaving only 8 ft. undeveloped. An appropriation of $7,000,000 has been made by the American Light & Power Co. for the completion of the additional improvements, already surveyed.

A great power loop is being built from Duluth, Minn., along the river, through Cloquet, to the heart of the iron-mining industry of Minnesota. This high-line will carry 110,000 volts and will connect up with all the different units.

The corporation will furnish light and power for Superior, Wisconsin, Duluth, Hibbing, Ely, Nashuak, and other range cities. Besides this, the lumber and paper manufacturing industries of Cloquet, Minn. will drive nearly all their machinery from the current, as will also the street-railway companies of Duluth and Superior.

Early in the summer the government granted the Weyerhaeuser interests and the Great Northern Power Co. the right to overflow several thousand acres of land in the northern part of the state so that the largest storage basin could be filled to capacity. This right was also transferred in the transaction.

Coal Storage Does Not Show Large Increase

According to the government report issued under the authority of the Federal Fuel Distributor, on July 1, 1923, commercial consumers had in storage approximately 45,000,000 net tons of bituminous coal. This was an increase over the revised figure of stocks on June 1, 1923, of 3,000,000 tons. Stocks on July 1 exceeded those on Aug. 1, 1921, by 4,000,000 tons and were nearly 5,000,000 tons larger than on April 1, 1919.

The stocks of anthracite in retail coal yards on July 1, 1923, though 13 per cent larger than on June 1, were much lower than during the spring and summer of preceding years. It has not been possible for the government to make a complete count, but a selected list of 418 dealers from whom reports have been received since 1919, shows total stocks of 787,426 tons on July 1, against 694,922 tons on June 1, 1923. These plants had 49 per cent less anthracite on hand on July 1 than on Aug. 1, 1921, and 17 per cent less than on Jan. 1, 1919. How much anthracite all dealers had is not known, but it is possible that the relation between stocks held by this group on different dates, might approximately represent all dealers.

There are no available statistics on householders' stocks of anthracite, but examination of production figures and stocks held by retailers indicate that householders remember their inability to procure a full supply of anthracite during last winter and are now laying in larger supplies than usual at this season of the year. The carry-over from last winter was very small, however, and it seems probable that the householders' bins are not now much more heavily stocked than is usual at this season.

It is of interest to note the statement made by the Massachusetts Fuel Administration that retail dealers in Massachusetts had 361,049 tons of anthracite on hand July 1, 1923.

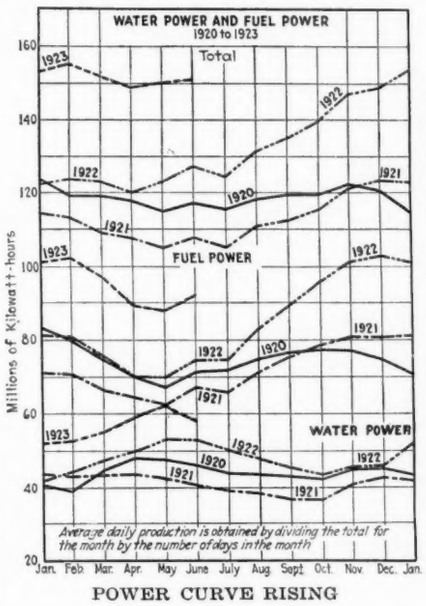
DOMESTIC ANTHRACITE COAL STOCKS IN MASSACHUSETTS, JULY 1, 1923
(NET TONS)

Stocks on hand June 1, 1923.....	181,887
Receipts April 1-June 30.....1,523,796	
Deliveries April 1-June 30.....1,344,634	
Stocks on hand July 1, 1923.....	361,049
Deliveries coal year ended March 31, 1922 5,243,415	

It seems evident, therefore, that householders in Massachusetts are taking their winter's supply almost as fast as it can be secured. It is not possible to state, however, how representative this condition is of the other anthracite-consuming states.

Production of Electricity Greatly Increased in 1923

According to the Geological Survey, there was produced, by public-utility



POWER CURVE RISING

power plants during the first half of 1923, 10,276,044,000 kw.-hr. of electricity by water power, and 17,186,236,000 kw.-hr. by fuels. There were 19,025,060 short tons of coal, 6,605,250 bbl. of fuel oil and 14,110,742 cu.ft. of natural gas consumed by these plants in producing this electricity.

The average daily production of electricity by public-utility power plants for the month of June was 151,000,000 kw.-hr.—practically the same as the revised figure for the May rate. An examination of the curve of total daily output for 1923, as shown by the chart, indicates that the increase in the demand for electricity at the present time is counterbalancing the usual seasonal decline in demand which usually occurs during the summer months. The average production of electricity during April, May and June apparently indicates that there will probably not be much change in the demand during the balance of the summer; the tendency is towards an increase rather than towards a decline in demand.

The average daily production of electricity for the first six months of 1923 and the proportion produced by water power were as follows: January, 153,300,000 kw.-hr.—34 per cent; February, 154,400,000 kw.-hr.—33.9 per cent; March, 152,500,000 kw.-hr.—36.3 per cent; April, 149,100,000 kw.-hr.—39.9 per cent; May, 150,100,000 kw.-hr.—41.3 per cent; and June, 151,000,000 kw.-hr.—38.9 per cent.

The average daily production of electricity during each of the past four years was as follows: 1919, 106,600,000 kw.-hr.; 1920, 119,000,000 kw.-hr.; 1921, 112,300,000 kw.-hr.; and 1922, 130,600,000 kw.-hr. During the first half of 1923 the average daily production of electricity was 151,700,000 kw.-hr.—an increase in rate of production of nearly 23 per cent over the average daily rate for the same period in 1922.

Water Diverted from Niagara River To Be Measured

The Canadian government has agreed to the plan proposed by the United States for the measurement of the diversion from the Niagara River. The measurements are to be made under the supervision of a board to be composed of two members, one to be appointed by the Canadian government and the other by the United States. The American member of the board will undoubtedly be Major P. S. Reinike, the district engineer of the War Department at Buffalo. Canada is expected to appoint William Stewart as its member of the board.

Nineteen thousand tons of Welsh coal were imported by Canada from Wales during June, 1923.

Obituary

John Hoyes McGowan, 93 years old and widely known as an inventor of pumps, and founder of the John H. McGowan Co., manufacturer of duplex steam and power driven pumps, died yesterday at his residence, Pleasant Ridge, Cincinnati, Ohio. Born in Aberdeen, Scotland, in 1830, Mr. McGowan came to America with his parents in 1832. They settled in Cincinnati in 1836. One of his seven surviving children, Mrs. Tony Sarg, is the wife of the New York cartoonist.

Hermann J. Strobel, supervisor of appraisals, of Stone & Webster, Inc. Boston, died Aug. 3 following an operation. He was 41 years old and had been with the Stone & Webster organization for the last eleven years in electrical-engineering work and ap-

praisal work. From 1905 to 1911 he was with the New York Central & Hudson River Railroad in New York as chief draftsman and assistant engineer on the electrical work of the Port Morris and Yonkers power stations and the Grand Central Terminal.

Personal Mention

M. W. Ward, formerly chief engineer of the Sims Co., has accepted a position as consulting engineer with the American Sterilizer Co., Erie, Pa.

J. E. Stark has been made vice-president of the Crane Company of Chicago, manufacturers of pipes and pipe fittings, and has gone to Paris, France, where his offices will be located.

Robert P. Guy has been appointed assistant chief inspector of the New York Branch of the Hartford Steam Boiler Inspection and Insurance Co. to fill the vacancy caused by the transfer of A. E. Bonnet.

E. Mason Parry, formerly in the Hartford office, has been appointed chief inspector of the New York Branch of the Hartford Steam Boiler Inspection and Insurance Co., to fill the vacancy caused by the death of Joseph H. McNeill.

L. M. Elliott has resigned his position as assistant superintendent of electrical construction of the Havana Electric Railway, Light & Power Co., to supervise the electrification of the sugar mills of the United Fruit Co., at Central Preston and Central Boston, Cuba.

Trade Catalogs

Stokers, Stowe — Laclede — Christy Clay Products Co., St. Louis Mo. Bulletins (2) describing in detail this stoker and its operation with other data and many illustrations.

Valves, Gate—Nelson Manufacturing Co., Philadelphia, Pa., Leafiets describing and illustrating gates valves Nos. 501-507, 509-510, 517-518, 520-522, 524-526, 528-530, 534.

Injectors — Penberthy Injector Co., Detroit, Mich. Directions for locating injector troubles gives data concerning the proper installation and care of injectors.

Lubricator — Keystone Lubricating Co., Philadelphia, Pa. Catalog consisting of 19 plates of illustrations with descriptive text, tables of dimensions and other data necessary for making a full presentation of this manifold safety lubricator.

Boilers—The D. Connelly Boiler Co., Cleveland, Ohio, Catalog handsomely illustrated, gives a description of water-tube boilers with drawings, diagrams and tables of properties of steam, carrying capacities of extra heavy steam pipes and height of chimneys in feet as well as many photos of installations of boilers.

Filters, Air—Midwest Air Filters, Inc., 100 East 45th St., New York City. Catalog, Dust Problems and Their Solution, illustrates with photos, drawings, tables and text the use and importance of this installation. A short history of air filters is found at the end of this handsome catalog of 49 pages.

Boiler Room Regulation—Hagen Corporation, Pittsburgh, Pa. This bulletin takes up the regulation and control apparatus manufactured by this company and gives definite recommendations in regard to the following installations: Control of the hand-fired plants; control of natural draft stoker-fired plants; control of forced-draft stoker-fired plants; control of forced-draft chain grate; control of powdered-coal firing apparatus and blast-furnace gas. Illustrations and drawings as well as the clear statements in the text make this bulletin useful to those interested in boiler and combustion control.

Coming Conventions

American Chemical Society, Dr. Charles L. Parsons, 1709 G St. N. W., Washington, D. C. Convention at Milwaukee, Wis., Sept. 10-14.

American Institute of Electrical Engineers, F. L. Hutchinson, 29 West 39th St., New York City. Pacific Coast Convention at Del Monte, Calif., Oct. 2-5.

American Institute of Mining and Metallurgical Engineers; F. F. Sharpless, 29 West 39th St., New York City. 128th annual meeting at Toronto, Canada, Aug. 20, Quebec, Aug. 27, Montreal, Aug. 30.

American Society of Mechanical Engineers, Calvin W. Rice, 29 West 39th St., New York City. Annual meeting at New York City, Dec. 3-6.

American Society of Refrigerating Engineers, William H. Ross, 35 Warren St., New York City. Nineteenth Annual Convention at New York City, Dec. 3-5.

Association of Iron & Steel Electrical Engineers; J. F. Kelly, 1007 Empire Bldg., Pittsburgh, Pa. Iron and Steel Exposition at Buffalo, N. Y., Sept. 24-28.

Empire State Gas & Electric Association, Charles H. B. Chapin, Grand Central Terminal Bldg., New York City. Meeting at Lake Placid, N. Y., Oct. 8-9.

Illuminating Engineering Society, H. E. Mahan, Schenectady, N. Y. Convention at Fort William Henry Hotel, Lake George, N. Y., Sept. 24-28.

Indiana Electric Light Association, T. Donahue, Northern Indiana Gas & Electric Co., Lafayette, Ind. Meeting at French Lick Springs, Ind., Sept. 26-29.

International Union of Steam and Operating Engineers; Dave Evans, 6334, Yale Ave., Chicago, Ill. Annual convention at Detroit, Sept. 10-15.

National Association of Practical Refrigerating Engineers; Ed. H. Fox, 914-25 East Jackson Blvd., Chicago. Fourteenth Annual Convention at Memphis, Dec. 12-16.

National Association of Stationary Engineers; Fred W. Raven, 417 South Dearborn St., Chicago, Ill. Annual convention and exhibition at Buffalo, N. Y., Sept. 10-15. Annual conventions and exhibitions of the State Associations scheduled as follows: **Pennsylvania**, at Buffalo, Sept. 9-10; J. N. Calvert, Crafton Sta., Pittsburgh, Pa. **New York**, at Buffalo, Sept. 9-10; W. T. Meinzer, Third St., near Warburton, Bayside, L. I., N. Y.

National Exposition of Power and Mechanical Engineering; Charles F. Roth, Room 1102, Grand Central Palace, New York City. Power Show, Dec. 3-8.

National Safety Council, W. H. Cameron, 168 North Michigan Ave., Chicago, Ill. Twelfth annual congress at the New Statler Hotel, Buffalo, N. Y., Oct. 1-5.

New England Association of Commercial Engineers; James F. Morgan, 53 Devonshire St., Boston, Mass. Power Show at Mechanics Bldg., Boston, Mass., Oct. 29-Nov. 3.

New England Water Works Association; Frank J. Gifford, 715 Tremont Temple, Boston, Mass. Annual Convention at Burlington, Vermont, Sept. 18-21.

Society of Naval Architects and Marine Engineers, Daniel H. Cox, 29 West 39th St., New York City. Annual meeting at New York City, Nov. 8-9.

West Virginia-Kentucky Association of Mine, Mechanical and Electrical Engineers; Herbert Smith, Room 211, Robson-Prichard Bldg., Huntington, W. Va. Third annual convention at Frederick Hotel, Huntington, W. Va., Oct. 19-20.

Fuel Prices

BITUMINOUS COAL

The following table shows the trend of the spot steam market in various coals (mine run bases, f.o.b. mines):

Coal	Market Quoting	Aug. 6, 1923	Aug. 13, 1923
Pool 1,	New York	$3.00@3.50	$3.00@3.50
Smokeless,	Columbus	2.75@3.25	2.75@3.25
Clearfield,	Boston	2.00@2.75	2.00@2.75
Somerset,	Boston	2.25@3.00	2.25@3.00
Kanawha,	Columbus	1.75@2.00	1.75@2.00
Hocking,	Columbus	1.75@2.00	1.75@2.00
Pittsburgh No. 8	Cleveland	1.85@2.00	2.00@2.10
Franklin, Ill.	Chicago	2.75@3.00	2.75@3.00
Central, Ill.	Chicago	2.00@2.25	2.00@2.25
Ind. 4th Vein,	Chicago	2.50@2.75	2.50@2.75
West Ky.,	Louisville	1.35@1.85	1.50@1.85
S. E. Ky.,	Louisville	1.50@2.00	1.50@2.00
Big Seam,	Birmingham	1.75@2.25	1.75@2.25

FUEL OIL

New York—Aug. 15, light oil, tank car lots, 28@34 deg. Baumé, 4c. per gal.; 36@40 deg. 4¼c. per gal., f.o.b. Bayonne, N. J.

Chicago — Aug. 6, 24@26 deg. Baumé, $1.97 per bbl.; 32@36 deg., $2.27 per bbl., tank cars.

St. Louis—Aug. 7, tank-car lots, f.o.b. St. Louis; 24@26 deg., $1.75 per bbl.; 26@28 deg., $1.80 per bbl.; 28@30 deg., $1.85 per bbl.; 32@36 deg., gas oil, 5c. per gal.; 36@40 deg., distillate, 5¼c. per gal.

Pittsburgh—Aug. 8, f.o.b. local refinery, 30@34 deg., fuel oil 4⅞c. per gal.; 36@40 deg., fuel oil, 5c. per gal.; 34 deg., neutral 7¼c. per gal.

Dallas—Aug. 11, f.o.b. local refinery, 26@30 deg., $1.39 per bbl.

Cincinnati—Aug. 14, tank car lots f.o.b. local refinery, 26@30 deg. Baumé 4¼c. per gal.; 30@32 deg., 4⅜c. per gal.; 38@42 deg., distillate, 6c. per gal.

Philadelphia—Aug. 9, 28 deg., 4.5@4.75c. per gal.; 30@34 deg., 5c. per gal.

New Plant Construction

PROPOSED WORK

Ark., Fort Smith—The city, c/o Clerk, is in the market for additional pump and electric motor for sewer pumping station on Wheeler Ave.

Calif., Delhi—The Delhi Land Settlement Colony is having plans prepared for the construction of three pumping plants, pumps about 12 in. diameter with motors 25 hp. or more, lift from water level about 45 ft. E. G. Fortier, c/o owner, Engr.

Calif., Helm — The Stinson Irrigation Dist., R. M. Bostwick, Secy., will receive bids until Sept. 4 for irrigation improvements as follows: 10 to 20 vertical shaft induction motors, 60 cycle, 1,200 r.p.m., 3 phase, 40 degree temperature rise, with starting compensator equipped with no voltage and overload relays, designed for conduit wiring, 20 and 25 hp. motor; 3 to 6 horiontal shaft induction, 20 and 25 hp. motors, 60 cycle, 3 phase, 860 r.p.m., 3 to 6 deep well turbine or propeller pumps, installed in 9⅝ in. I. D. wells, 60 ft. column, 20 suction, to pump from 2 to 2.5 second feed with a total lift of from 35 to 50 ft., direct connected to vertical shaft motors, 1,200 r.p.m. or belt driven by horizontal motors, 860 r.p.m.; 10 to 14 deep well turbine or propeller pumps, installed in 14 in. wells, 60 ft. column, 20 ft. suction, to pump from 2 to 3 second ft., total lift from 35 to 50 ft., direct connected to vertical shaft motors, 1,200 r.p.m.; drilling 3 to 6 wells, 400 to 800 ft. deep; 14 to 20 pump houses, derricks, etc.; instaling 15 to 20 deep well pumps and motors with piping; about 148 metal gates, 20 in. diameter; about 20 head gates, in canals; 15 syphons across sloughs. Quinton, Code & Hill, 1106 Hollingsworth Bldg., Los Angeles, Consult. Engrs.

Calif., Long Beach—The city is having plans prepared for the construction of a memorial auditorium. Estimated cost $2,000,000. Wright & Gentry, Marine Bank Bldg., Los Angeles, Archts. Equipment detail not reported.

Calif. Los Angeles — The Federal Refrigerating Co. is having preliminary sketches prepared for the construction of a 6 story, 200 x 400 ft. ice and cold storage plant on Downey Rd. Estimated cost $2,000,000. F. D. Chase, Inc., 533 Title Insurance Bldg., Los Angeles, Archt.

Calif., Marysville—The City Trustees, J. O. Wanzer, Mgr., appropriated $3,500 for the purchase of new pumps and switches for city sump.

Calif., Modesto—The Modesto Irrigation Dist., C. S. Abbott, Secy., will receive bids until Aug. 30 for furnishing transformers and meters for 12 month contract. Estimated cost $30,000 for transformers and $5,000 for meters.

Calif., Oakland—M. I. Diggs, owner and Architect, 1901 Telegraph Ave., is preparing plans for the construction of an 8 story store, office and theater building on 19th St. and Telegraph Ave. Estimated cost $500,000. Equipment detail not reported.

Calif., Oakland—The Pacific Gas & Electric Co., 445 Sutter St., San Francisco, is having plans prepared for the construction of a 2 story, 75 x 100 ft. substation on 21st St., here. Estimated cost $280,000. Private plans.

Calif., San Diego—The Army and Navy Y. M. C. A. had plans prepared and will receive bids until Aug. 25 for the construction of a Y. M. C. A. building, including 50 x 80 ft. gymnasium, cold storage rooms, etc., with two elevators, on Broadway. L. Rogers, Spreckels Bldg., Los Angeles, Archt. Noted Nov. 21, 1922.

Calif., San Francisco—The city and county are having plans prepared for the construction of a power plant for distribution of power generated from Hetch Hetchy project. M. M. O'Shaughnessy, Engr., City Hall.

Calif., San Francisco—The Eng. Dept. of the Pacific Gas & Electric Co., 445 Sutter St., is receiving bids for steel for the construction of a 1 story, 75 x 100 ft. steam plant, also equipment, on Stevenson St., between 5th and 6th Sts. Estimated cost $250,000. Private plans. Noted July 24.

Calif., San Francisco—The Pacific Gas & Electric Co., 445 Sutter St., is having plans prepared for an electric substation. Cost will exceed $50,000. Private plans.

Calif., San Jose—The city received bids for the construction of a power plant on the county hospital grounds from D. J. Byron, 1249 Alameda St., San Jose, $18,587; R. O. Summers, 17 North 1st St., San Jose, $18,731; Jorgenson & Cook, 651 Provost St., San Jose, $18,969; mechanical equipment from J. E. O'Mara, 218 Clara St., San Francisco, $86,761; J. Nelson, 517 6th St., San Francisco, $87,393; Scott Co., 243 Minna St., San Francisco, $96,500. Noted Aug. 7.

Calif., Stockton—The College of the Pacific, c/o Davis-Heller-Pearce Co., Archts., Delta Bldg., is having plans prepared and will receive bids after Sept. 1 for the construction of seven 2 story college buildings. Estimated cost $600,000. Architects as follows: Losekann & Clowsley, Elks Bldg., Stockton, for science building; Stoner & Warner, Belding Bldg., Stockton, for dining hall and gymnasium; Mayo, Cowell & Bissell, 24 South San Joaquin St., Stockton, for auditorium; Davis-Heller Pearce Co., Delta Bldg., for boys' and girls' dormitory, power house and stadium. The latter architects will act as managers of construction for the entire project. Equipment detail not reported. Noted Jan. 9.

Ill., Chicago—T. R. Bishop, Archt., 35 South Dearborn St., is receiving bids for the construction of a 5 story, 150 x 150 ft. apartment hotel, including steam heating system, at 5474-84 Cornell Ave. for G. G. & E. O. Anden, c/o Archt. Estimated cost $1,000,000.

Ill., Chicago—Gumbinsky Bros. Co., 2261 South Union Ave., is having plans prepared for the construction of a group of paper mill buildings on a 20 acre site on 31st and California Sts. Estimated cost $4,000,000, including $1,500,000 worth of equipment. A. Epstein, 2007 Pershing Rd., Chicago, Engr. B. M. Barter, Grand Rapids, Mich., Mech. Engr.

Ill., Chicago—A. F. Himelblau, Archt., 30 North Dearborn St., is receiving bids for the construction of a 4 story, 120 x 250 ft. apartment house, including steam heating system, on Howard and Marshfield Sts. for F. H. Janiszeske, 1373 West Chicago Ave. Estimated cost $500,000.

Ill., Chicago—A. Ingram, c/o R. Huszaah, 105 North Clark St., is having plans prepared for the construction of a 3 story, 53 x 282 ft. apartment house, including steam heating system, on Eastlake Terrace and Rogers Ave. Estimated cost $500,000.

Ill., Chicago—The Neptune Linen Supply Co., 8056 39th St., is in the market for power plant equipment.

Ill., Chicago—Shattuck & Layer, Archts., 19 South La Salle St., will soon receive bids for the construction of a 3 and 4 story clubhouse, including steam heating system, on Madison St. near Cicero St. for the Columbus West Side Bldg., Corp., c/o Architect. Estimated cost $700,000.

Ill., Freeport—The W. T. Rawleigh Co., 101 South Liberty Ave., is having plans prepared for the construction of a 1 story, 100 x 110 ft. power plant. Estimated cost $200,000. J. R. and E. J. Law, 1st Central Bldg., Madison, Archts.

Ill., Kewanee—The city, R. R. Haley, Clk., will receive bids until Aug. 28 for improvements to electric light plant (1) one 300 kw. and one 500 kw. turbo generator with condensors and switchboard; (2) one 350 hp. water tube boiler, also breeching and mechanical stoker; (4) construction of addition to building, machinery foundations and spray pond; (5) street lighting system. Burns & McDonnell Eng. Co., 402 Interstate Bldg., Kansas City, Mo. Noted Feb. 6.

Ill., Pittsfield—The city will receive bids Sept. 1 for the construction of an earth dam, concrete gate chamber, concrete conduit, spillway, complete filter plant and pumping plant, including pumping machinery and force main to the city. The work is divided into five different divisions with separate items. The city will receive bids direct on furnishing of pumping equipment and c.i. pipe. Caldwell Eng. Co., Jacksonville, Engrs.

Ind., Columbia City—The city will soon award the contract for the construction of an addition to light and waterworks plant, also for installation of boilers, stokers, soot blowers and superheaters. Estimated cost $30,000. C. Brossman, 1503 Merchants Bank Bldg., Indianapolis, Consult. Engr.

Ind., Muncie—The Bd. of Trustees, Indiana State Normal School, H. C. Benbridge, Secy., will soon award the contract for the construction of a power house and tunnel chimney, including boilers, stokers and soot cleaners. Estimated cost $60,000. Kibele & Garrard, 335 Johnson Bldg., Muncie, Archts.

Kan., Lyons—The city, S. L. Dinsmore, Clk., will receive bids until Aug. 27 for c.i. pipe, manholes, flush tanks, etc.; Imhoff tank; four intermittent filters; smudge drying bed; one 200 g.p.m. duplex ejector with motors, compressors; one 100 g.p.m. ejector; one air tank, etc. Estimated cost $90,000. C. A. Haskins, 517 Finance Bldg., Kansas City, Engr.

Ky., Danville—The City Council is in the market for electrically operated pumps for water plant. Peare, Greeley & Hansen, 38 West Adams St., Chicago, Engrs.

Ky., Louisville—The Bd. Educ. plans an election in November to vote on $1,000,000 bonds for the construction of new school buildings.

La., New Orleans — The Presbyterian Hospital, 719 Carondelet St., is having plans prepared for the construction of a 10 story hospital on Baronne and Girod Sts. Estimated cost $1,250,000. DeBuys & Armstront, Hibernia Bldg., New Orleans, Archts. Equipment detail not reported.

La., New Orleans—The Saenger Amusement Co., Tulane and Liberty Sts., is having plans prepared for the construction of a theater on Canal and North Rampart Sts. Estimated cost $1,000,000. E. Weil, Whitney Annex, Archt. Equipment detail not reported.

Md., Baltimore—Beckerley & Trusler, 614 East Lombard St. (marine supplies, machinery, etc.), C. Trusler, Purch. Agt., are in the market for centrifugal pumps, up to 4 in.

Mass., Boston—The city, City Hall, is having plans prepared for the construction of a 4 story high school on East Eagle St. Estimated cost $1,000,000. J. Gray, 175 High St., Boston, Archt. Equipment detail not reported.

Mich., Big Rapids — The Big Rapids Water Power Co. plans the installation of generating equipment and spillway improvements. Burd & Giffels, Powers Theatre Bldg., Grand Rapids, Engrs.

Mich., Ithaca—The city plans the installation of deep well plunger pumps and a fire pump unit. Estimated cost $11,000. Burd & Giffels, Powers Theatre Bldg., Grand Rapids, Engrs.

Mo., Norborne—The city plans the construction of a pumping plant, reservoir, tower and distribution mains. Cost between $50,000 and $60,000. Engineer or architect not selected.

Mo., St. James—The city voted $45,000 bonds, $33,000 for waterworks pumping plant and distribution system, $12,000 for sewer system. Engineer or architect not selected. Noted July 24.

Mo., Stockton—The city will soon receive new bids for the construction of a 1 story, 25 x 40 ft. power house, spillway, 130 ft. long, 10 ft. high; two 36 in. vertical reacting turbines; two 2,300 volt cycle generators; 18 mi. transmission line from here to Fairplay. Former bids rejected. Estimated cost $60,000., Private plans. Noted Aug. 7.

N. Y., Brooklyn—The St. George Hotel, Henry and Clark Sts., is having plans prepared for the construction of a 12 story addition to hotel. E. Roth, 119 West 40th St., New York, Engr. and Archt. Equipment detail not reported.

N. Y., New York—The Bd. Educ., c/o W. H. Gompert, Archt., Flatbush Ave. Extension and Concord St., Brooklyn, received bids for heating and ventilating apparatus for Julia Richman High School at 2nd Ave. and 67th St., here, from Gillis & Geoghegan, 537 West Broadway, $268,630; E. Rutzler Co., 404 East 49th St., $273,459; W. J. Olvaney, Inc., 100 Charles St., $277,000. Noted Aug. 7.

N. Y., New York—The Park Bd., F. D. Gallatin, Pres., Dept. of Parks, 10th floor, Municipal Bldg., will receive bids until Aug. 23 for alterations and additions to electric generating plant of the Amer. Museum of Natural History, Manhattan Sq.

N. D., Dunseith—The State Bd. of Administration, Bismark, E. G. Wanner, Secy., will receive bids until Aug. 28 for the construction of a 1 story power house and tunnel, also ash conveyor, at the State Tuberculosis Sanatorium, here. Estimated cost $40,000. Keith & Kurke, Equity Bldg., Fargo, Engrs.

Ohio, Sandusky — M. Moskowitz, 2179 East 83rd St., Cleveland, representing owner, is having plans prepared for the construction of an 8 story, 130 x 130 ft. hotel, including steam heating system, here. Estimated cost $1,000,000. J. H. Graham & Co., 8201 Cedar Ave., Cleveland, Archts.

Ohio, Springfield—The County Comrs. are having plans prepared for the construction of a tuberculosis hospital, consisting of five buildings. Estimated cost $275,000. L. W. Hall, c/o Hall & Lethly, 806 Fairbanks St., Springfield, Archt.

Ohio, Warren—The Warren Hotel Co., c/o P. E. Robinson, 627 Bulkley Bldg., has had plans prepared for the construction of a 12 story, 81 x 176 ft. hotel, including steam heating system. Estimated cost $1,000,000. Crosby Eng. Co., Bulkley Bldg., Archts.

Okla., Picher—The Oklahoma Mining Co., A. Reno, Purch. Agt. plans to electrify plant and is in the market for motors, hoists, etc.

Ore., Portland—The Public Dock Comn., J. H. Polhemus, Genl. Mgr., 916 Spaulding Bldg., is having plans prepared and will soon receive bids for the construction of a refrigeration plant and storage room at Municipal Dock 4. G. B. Hegardt, Stark St. Dock, Ch. Engr.

Ore., Stayton—A. D. Gardner has been granted permit by state engineer to appropriate 1,000 second ft. of water from the North Fork of the Santiam River at four points between here and Mehama, to develop 13,636 hp. Estimated cost $250,000. Engineer not announced.

S. D., Watertown—The Municipal Utilities Comn., A. M. Myhre, Secy., will receive bids until Aug. 29 for supplying, installing and completing filter, pumping and piping equipment for waterworks filter house and pumping station. W. F. Cochrane, Engr.

Tenn., Memphis—The Snowden Estate is having plans prepared for the construction of an 8 story, 148 x 148 ft. department store, including 8 elevators and heating system, on South Main St. and Monroe Ave. Estimated cost $1,250,000. Hanker & Cairns, Scimitar Bldg., Memphis, and E. Weil, Whitney Annex, New Orleans, La., Archts. B. Lowenstein Bros., Inc., Main and Court Sts., Memphis, lessee

Tex., Dallas — Koch & Fowler, Engrs., Central State Bank, are receiving bids for a steel tank and tower or concrete tank and pumping unit near Midway Rd. for J. P. Stephenson, c/o Engrs. Cost between $15,000 and $20,000.

Tex., Mart—The city is having plans prepared for the construction of a reservoir, pumping station and c.i. mains. Estimated cost $200,000. Koch & Fowler, Central Bank Bldg., Dallas, Engr.

Wis., Green Bay—Platten Bros. Co., 413 Dousman St., is in the market for a 4 ton ice machine, refrigeration machinery, etc.

Wis., Hartford—The Spaeth Packing Co. plans to rebuild its 1 story, 32 x 250 ft. packing plant, which was destroyed by fire. Estimated cost $50,000. Engineer or architect not selected. The owner is in the market for motors, etc.

Wis., Iron River—The Iron River Water & Light Co., F. F. Macmiller, Pres., is receiving bids for the construction of a dam, also generator for power plant. Private plans. Noted Jan. 2.

Wis., Madison—The Bd. of Regents, University of Wisconsin, Administration Bldg., J. D. Phillips, Purch. Agt., is in the market for refrigeration equipment and machinery for new state hospital.

Wis., Madison — The Chicago & Northwestern Ry., c/o B. R. Kulp, 2015 South Blair St., is receiving bids for the construction of a 1 story, 50 x 104 ft. power house. Estimated cost $100,000. The owner is in the market for power machinery, motors, etc. Noted June 5.

Wis., Madison—The Security State Bank, c/o L. Lumenschloss, 629 Atwood Ave., is receiving bids for a steam heating plant, boilers, radiation and piping for bank building. Estimated cost $25,000. F. Riley, Conklin Bldg., Madison, Engr.

Wis., Menasha—J. Herzizer, Main St., is receiving bids for a steam heating plant, including boilers, radiation and piping. Estimated cost $25,000. E. A. Wettengel, Appleton, Engr.

Wis., Milwaukee—The Ku Klux Klan, c/o W. F. Wieseman, 1037 39th St., plans to build a lodge building. Estimated cost $500,000. Engineer or architect not selected.

Wis., Milwaukee—Marquette University, H. Banzhaf, Business Mgr., 1115 Grand Ave., plans to build additional buildings, consisting of medical building, hospital, nurses' home and an addition to dental building. Estimated cost $500,000. Engineer or architect not selected.

Wis., Milwaukee—The Sewerage Comn., 508 Market St., J. H. Fowles, Secy., will receive bids until Aug. 24 for high pressure steam piping for boiler and power house on Jones Island. T. C. Hatton, c/o owner, Engr.

Wis., Milwaukee—P. Verhalen, 1129 35th St. (meats, etc.), is in the market for a 4 ton ice making machine, motor power.

Wis., Montfort — The city, c/o Clerk, plans election to vote on $15,000 bonds for watermain extension and additional pumping equipment. Engineer not selected.

Wis., Phillips—The city has had plans prepared and will soon receive bids for a 100,000 gal. steel tank on 50 ft. tower, 1,000,000 gal. reservoir, two pumping engines, c.i. water pipe and mains. Estimated cost $35,000. F. Henry, Courthouse, Engr. Noted Mar. 27.

Every one of these items is reported by our authorized correspondents who are instructed to verify every item sent in. This free weekly service is published in the interests of the buyer and the seller, to bring them together and get machinery moving.

Everything possible is done to insure authenticity and timeliness.

Your co-operation is invited in helping us maintain this service at the highest effic ency.

BUSINESS NEWS DEPARTMENT
Tenth Ave. at 36th St., New York

Wis., Sheboygan—The United States Coast Guard, G. W. Kruelle, Supt., is having plans prepared for the construction of a steam heating plant, including boilers, for Coast Guard Station. Private plans.

Wis., Sheboygan Falls — The city, C. Herman, Clk., will receive bids until Aug. 29, for the construction of a reservoir, pumping station and installation of water mains. Estimated cost $30,000. J. Donohue & Co., 608 North 8th St., Sheboygan, Engr. Noted Feb. 13.

Alta., Drumheller—The town is in the market for two electrically driven sewerage pumping units with accessories, including pumps, capacity 300 g.p.m., vertical shaft type; 5 hp., 200 volt, 3 phase, 60 cycle motors, 40 degree C induction; float switches and control panel.

Que., Farnham—The Ice Mfg. Co., Ltd., is in the market for ice manufacturing machinery and equipment.

Que., Montreal—The Montreal Light, Heat & Power Co., Craig St., W., C. S. Bagg, Secy., is in the market for one electrically driven coal crusher for steam station at Lasalle, also one electric boiler, about 300 hp. Estimated cost $3,000, each.

CONTRACTS AWARDED

Ark., Forrest City—The city awarded the contract for the construction of a power plant, including traveling crane, to C. A. Ahlsteadt, Forrest City. Estimated cost $10,125. The owner will purchase equipment, including engines, motors, etc., E. W. Mann, Memphis, Tenn., Engr. Noted July 31.

Calif., Lodi—The Gloria Ice Cream Co., Oakland and Aurora Sts., will build the first unit of ice cream factory, consisting of a refrigerator plant, on Central and Railroad Aves. Work will be done by separate contracts under supervision of J. F. Watson, Archt., 403 East Pine St. Estimated cost $50,000. Noted June 19.

Ill., Chicago—S. J. Robbins, 140 North Dearborn St., representing owner, awarded the contract for the construction of a 2 story, 156 x 225 x 350 ft. store, office and dance hall on Milwaukee and Lyndale Sts. to McKeown Bros., 102 West Adams St., Chicago. Estimated cost $1,250,000. Steam heating system will be installed.

Minn., Lesueur Center—The city awarded the contract for a motor driven, deep well pumping unit and pipe fittings to connect with street main to J. G. Robertson, 2542 University Ave., St. Paul. Estimated cost $3,475. Noted July 24.

Minn., Minneapolis—H. S. Goldie, 910 Newton Ave., representing owners, awarded the contract for the foundation of a 6 story, 120 x 250 ft. apartment hotel on West Lake St. to the Standard Home Bldg. Co., Andrus Bldg., Minneapolis. Bids for superstructure to be asked later. Estimated cost $1,300,000. This corrects report in July 10th issue.

Okla., Tulsa—The city awarded the contract for equipment for Mohawk pumping station, two 12 m.g.d. high service steam turbine driven centrifugal pumps to the DeLaval Steam Turbine Co., North Clinton St., along tracks of Pennsylvania R.R., Trenton, N. J., $84,300; two 300 hp. water tube boilers with Green chain stokers to the Walsh & Weidner Boiler Co., Main and Sycamore Sts., Chattanooga, Tenn., $24,215. Contract for construction of pumping station will be awarded about Oct. 1. Noted July 17.

Pa., Harrisburg—The Penn-Harris Hotel Co., 3rd and Walnut Sts., awarded the contract for the construction of a 12 story, 30 x 200 ft. addition to hotel to the Hunkin-Conkey Constr. Co., New Century Bldg., Cleveland, Ohio. Estimated cost $800,000. Steam heating system will be installed. Noted Feb. 13.

Pa., Phila.—The Colonial Knitting Mills Inc., Clearfield, St., awarded the contract for the construction of a 3 story, 86 x 170 ft. hoisery mill and a 1 story, 45 x 110 ft. boiler and dye house to the Austin Co., Jefferson Bldg., Phila. Estimated cost $245,000. Elevators, boilers, engine and dyehouse equipment, also heating and ventilating system will be installed.

Tex., Corsicana—The Navarre Ice Co. will build a 1 story, 90 x 150 ft. ice plant, 100 ton capacity, electric or oil engine driven, raw water system, on 12th St. along tracks of Cotton Belt R. R. Estimated cost $100,000. Noted Aug. 14.

Tex., Denison—The Denison Hotel Co. will build a 6 story, 110 x 120 ft. hotel. Estimated cost $250,000. Equipment detail not reported. Noted July 10.

Tex., Fort Worth—The city awarded the contract for 16 in. Worthington Class B S, 2 stage horizontal split case, double suction, volute pumps, made up of two 16 in. 500 g.p.m against 45 ft. head for Holly pumping plant to A. M. Lockett & Co., Magnolia Bldg., Dallas. Estimated cost $20,993. Noted June 19.

Wash., Seattle—The State Capitol Comn. awarded the contract for the construction of a capitol and legislature building to the Sound Constr. & Eng. Co., Lownan Bldg., Seattle. Estimated cost $780,000. Equipment detail not reported.

Wis., Manitowoc—The city, A. Zander, Clk., awarded the contract for the construction of a 1 story, 55 x 65 ft. water and light plant to T. Stark & Co., 130 Muskego Ave., Milwaukee. Noted July 17.

Wis., Marshfield — The city, A. L. Brownell, Supt., awarded the contract for additional water supply, capacity 2,000,000 g.p.d., including construction of pump house, equipment, piping, wells and mains, to the Layne & Bowler Co., 37 West Van Buren St., Chicago. Estimated cost $160,000.

Wis., Thorp—The city, E. W. Wolf, Pres., awarded the contract for the construction of sewer extensions and disposal plant to E. C. Manson, Withee. W. G. Kirchoffe, Dean Bldg., Madison, Engr., is receiving bids for a compressor and motor. Noted July 17.